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ENERGY CONSERVATION: HEATING NAVY HANGARS

J. L. ASHLEY, E. CORREA, K. CANFIELD

JULY 1984

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NAVAL CIVIL ENGINEERING LABORATORY
PORT HUENEME, CALIFORNIA 93043

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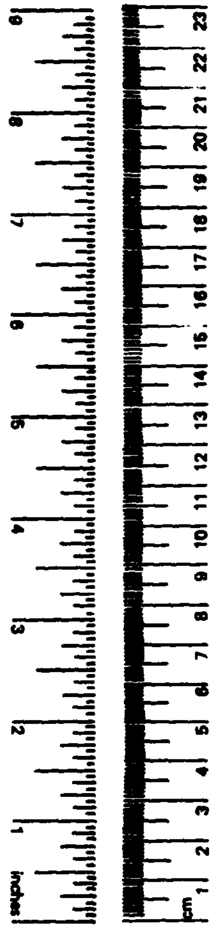
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

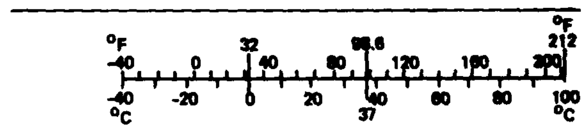
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2,000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1,000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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How energy is used for hangar heating and what methods are used to reduce hangar thermal energy consumption were investigated. The results of measurements of hangar air infiltration and stratification, two major causes of heating-related energy consumption, are reported. Methods to reduce this type of energy consumption (reduction of air infiltration and installation of destratifiers, vehicle access doors, door seals, vinyl strip doors, and radiant heating) were evaluated and are discussed. Design criteria providing hangar air infiltration

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rates versus hangar size and climatic conditions and design criteria for hangar destratifiers were developed and are presented. Energy Monitoring and Control Systems (EMCS) for saving cooling and heating energy are discussed.

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INTRODUCTION

The U.S. military has airfields and hangars located in cold weather regions in many parts of the world. Hangars, because of their large interior volumes, large door-to-wall-area ratios, and their open interiors, consume large quantities of energy for space heating. Since hangar heating can represent a significant percentage of the heating-related energy consumed at an air facility, the Naval Facilities Engineering Command (NAVFAC) tasked the Naval Civil Engineering Laboratory (NCEL) with investigation of methods and construction techniques to reduce energy requirements, primarily for hangar heating. The Engineering Service Command, Tyndall Air Force Base, Fla., joined in the effort and supported the investigation. Assistance was also provided by the following Naval and Air Force Commands: Naval Air Rework Facility, Norfolk, Va.; Naval Air Development Center, Warminster, Pa.; Naval Air Station, Brunswick, Maine; Minot Air Force Base, Minot, N.D.; and McClellan Air Force Base, Sacramento, Calif.

RESULTS OF ENERGY LOSS INVESTIGATION

Aircraft hangar space heating has always been difficult because of air stratification, air infiltration, slow heat recovery, and high heating costs. Air infiltration and stratification have been determined as the major causes of high energy consumption in hangars.

When forced-air convective heaters are used, the warm air rises, resulting in high air temperatures near the ceiling and lower temperatures at the floor level. This stratification results in excessive energy consumption. In addition, when the main doors are opened for aircraft passage, heated air is lost to the colder outside environment. When the doors are closed, the heating process must start all over again.

As part of the work reported in this document, hangar heating characteristics were measured. Reference 1 presents the air infiltration and stratification measurements made at four military hangars. As a result of those measurements, evaluation of proposed energy conservation concepts was begun. Several hangar heating-related energy conservation concepts have been evaluated since the 1982 report, and their results are presented in this document. The exterior surface insulation requirements for hangars is similar to other industrial-type structures and was not part of this investigation.

AIR INFILTRATION

The entrance of unwanted cold outside air into a heated structure is known as air infiltration. Each cubic foot of cold outside air that enters a heated structure displaces an equal amount of warm air and must, in turn, be heated. In this report, the energy required to heat the cold outside air is considered air infiltration loss.

Air infiltration is dependent upon the pressure difference between the inside and outside of a structure and upon the area of openings in a structure's skin. The inside/outside pressure difference is, in turn, dependent upon the inside/outside air temperature difference, the height of the structure, wind velocity, and mechanical ventilation. Hangars typically utilize natural ventilation techniques; thus, mechanical ventilation is not a factor and does not require consideration. The inside/outside air temperature difference and the structure's height are coupled together and are the dominant cause of air infiltration when wind speeds are low (less than 5 mph). At moderate or high wind speeds, the wind dominates the inside/outside pressure difference.

Recent investigations at NCEL and Lawrence Berkeley Laboratory of building air infiltration (Ref 2 through 5) have determined that air infiltration can be expressed as a function of wind speed, inside/outside air temperature difference, crack area, and structure height by the following relationship:

$$Q = \frac{I\ V}{60} = A_o\ [(f_w)^2\ W^2 + (f_s)^2\ \Delta T]^{1/2} \quad (1)$$

where: Q = air infiltration, ft³/min

I = air changes, no./hr

V = building volume, ft³

A_o = total crack area, ft²

f_s = stack effect parameter, defined by Equation (4)

f_w = wind effect parameter, defined by Equation (7)

ΔT = inside/outside air temperature difference, °F

W = wind speed, ft/min

The horizontal fraction of the total leakage, R (leakage from horizontal cracks), is assumed to be 0.3 for buildings such as hangars constructed on concrete slab foundations.

$$R = \frac{\text{ceiling leakage + floor leakage}}{\text{total leakage}} = 0.3 \quad (2)$$

If the fractional difference, X, between the ceiling and floor leakage is assumed to be negligible, then

$$X = \frac{\text{ceiling leakage - floor leakage}}{\text{total leakage}} \cong 0 \quad (3)$$

The stack effect parameter, f_s, can be expressed as:

$$f_s = \frac{1}{3}\left(1 + \frac{R}{2}\right) \left[1 - \frac{X^2}{(2 - R)^2}\right]^{3/2} \left(\frac{g\ H}{T}\right)^{1/2} \quad (4)$$

where: T = interior air temperature, °R

H = interior height, ft

g = acceleration of gravity, ft/min²

Substituting for R, X, and g

$$f_s = 130.5 \left(\frac{H}{T}\right)^{1/2} \quad (5)$$

Assuming an average inside temperature of 525°R, f_s becomes

$$f_s = 5.7\sqrt{H} \tag{6}$$

The wind effect parameter, f_w, is expressed as:

$$f_w = C(1 - R)^{1/3} [K H^{\alpha}] \tag{7}$$

where C is a shielding coefficient and K and α are terrain coefficients. The shielding coefficient is determined by trees, fences, vehicles, and other structures in the immediate vicinity of the hangar. The terrain coefficients are dependent upon large-scale obstructions within several miles of the hangar and are experimentally determined. Typically, at a military air facility, aircraft, vehicles, hangars, and other structures are usually nearby, but a large open area containing the air strip must also be considered. For these conditions, the shielding coefficient is associated with light urban shielding, and the terrain coefficients are associated with urban- or industrial-type terrain. From Tables 1 and 2, the following values of C, K, and α were assumed:

$C = 0.285$
$K = 0.40$
$\alpha = 0.25$

Thus, $f_w = 8.91H^{1/4}$ (8)

Table 1. Terrain Coefficients Affecting Wind in Vicinity of Hangar

Terrain Coefficients		Description
K	α	
0.68	0.17	Flat
0.52	0.20	Rural
0.40	0.25	Urban
0.31	0.33	City

Table 2. Shielding Coefficients Affecting Wind in Vicinity of Hangar

Shielding Coefficient C	Description
0.285	Light
0.240	Moderate
0.185	Heavy
0.102	Very Heavy

Substituting f_s and f_w into Equation 1 and expressing wind speed in units of miles per hour,

$$Q = \frac{I\ V}{60} = A_o\sqrt{79.34H^{1/2}\ S^2 + 32.43H\ \Delta T} \qquad (9)$$

where S is wind speed in miles per hour.*

Measurements were made to determine typical air infiltration rates for military hangars. The tracer gas dilution method (ASTM E741-80) was used to measure actual leakage for the following hangars:

Activity	Hangar Number
Naval Air Rework Center, Norfolk, Va.	V-147 and V-88
Naval Air Station, Brunswick, Maine	250
Minot Air Force Base, Minot, N.D.	867
McClellan Air Force Base, Sacramento, Calif.	365

Sulfur hexafluoride (SF_6) was used for the tracer gas, and a gas chromatograph was used to measure tracer gas decay (Ref 6). The results of the measurements are presented in Figures 1 through 10. The total crack area was calculated for each hangar, and the ratio of the crack area to the total wall surface area was determined (Table 3).

A ratio (expressed as a percentage) of crack area to total vertical surface area of 0.23% was typical for three of the five hangars measured. The increased crack-to-surface area percentage for the remaining two hangars resulted from deteriorated or missing hangar door seals. While none of the hangars measured were without deficiencies, the three hangars with a ratio of crack area to vertical surface area of 0.23% were in good maintenance condition and are typical of most military hangars.

If a total crack area to vertical surface area percentage of 0.23 is assumed to be typical of a hangar in good maintenance condition, then Equation 9 can be modified to calculate the air change ratio for use in developing design criteria for a heating system. By substituting the following,

$$A_o = \frac{0.23A_w}{100}$$

where: A_w = total vertical surface area

*The numeric constant 79.34 has been adjusted to convert wind speed from miles per hour to feet per minute.

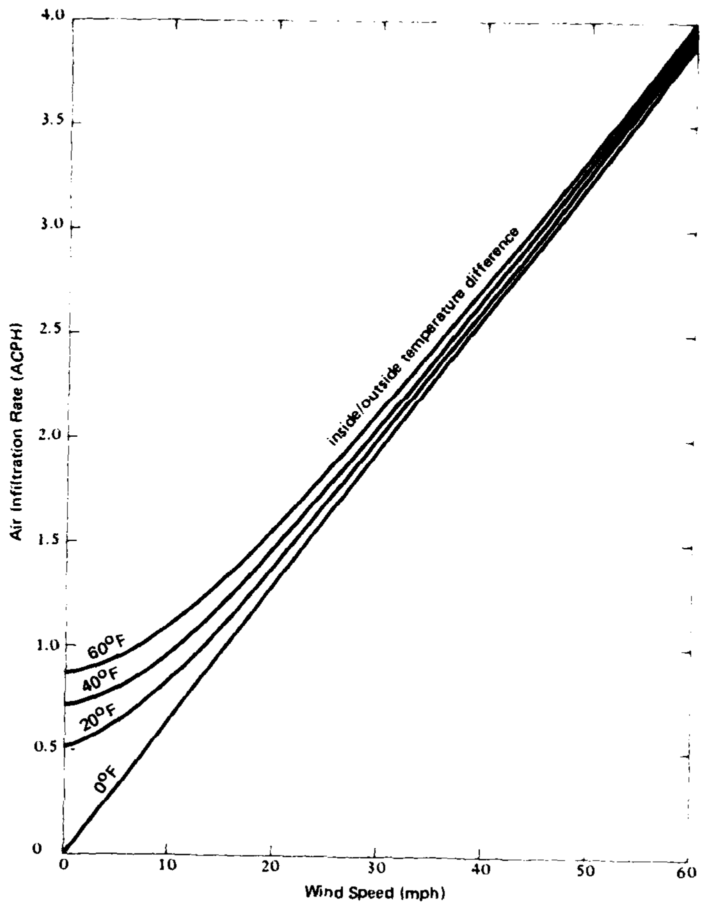


Figure 1. Air infiltration rate versus wind speed (doors closed) for Hangar V-147, NARF Norfolk.

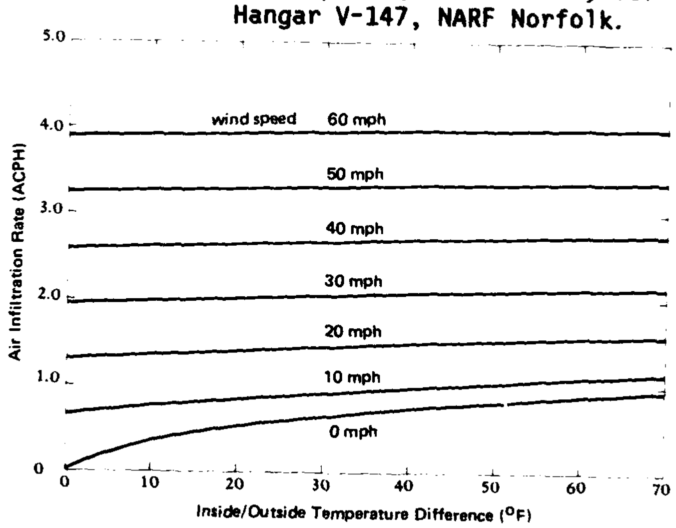


Figure 2. Air infiltration rate versus temperature difference (doors closed) for Hangar V-147, NARF Norfolk.

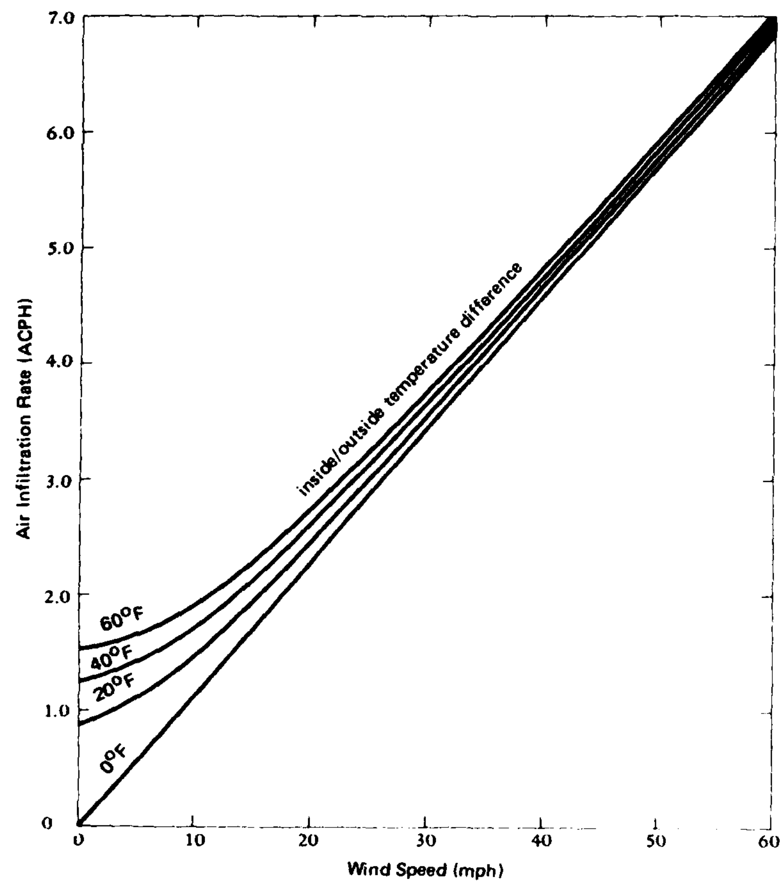


Figure 3. Theoretical air infiltration rate versus wind speed (doors closed) for Hangar 867, Minot AFB.

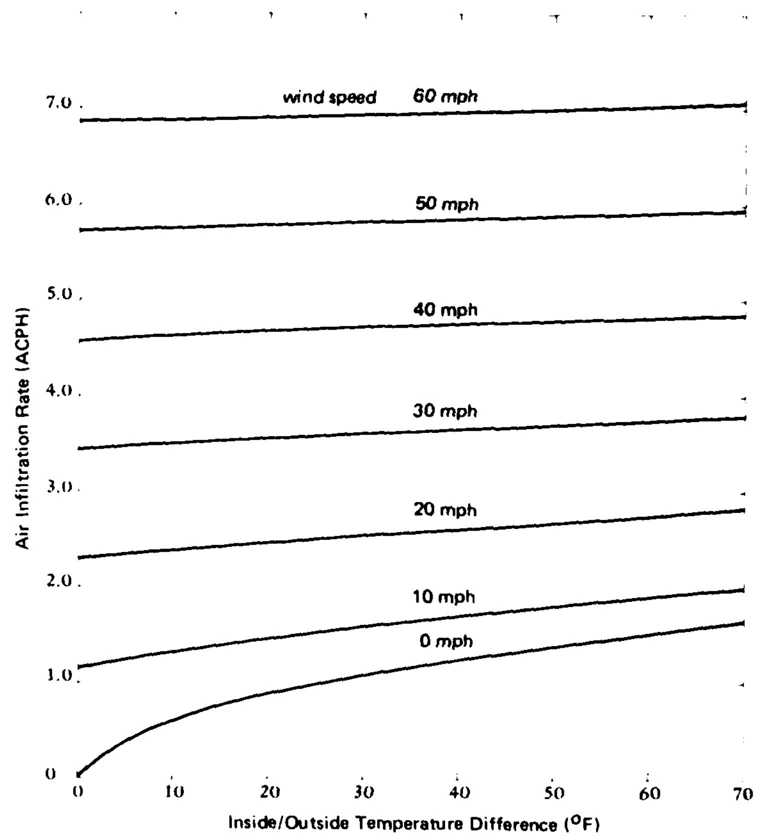


Figure 4. Theoretical air infiltration rate versus temperature differential (door closed) for Hangar 867, Minot AFB.

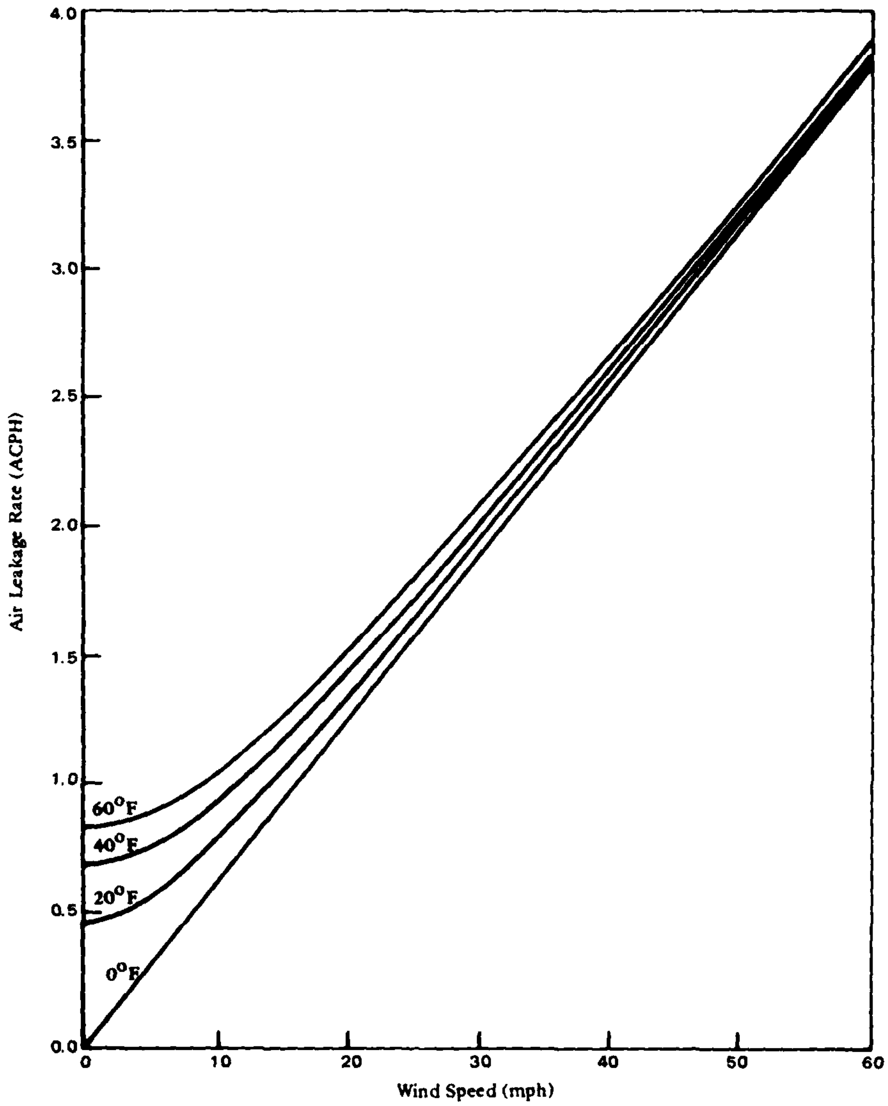


Figure 5. Air infiltration rate versus wind speed for Hangar V-88, NARF Norfolk.

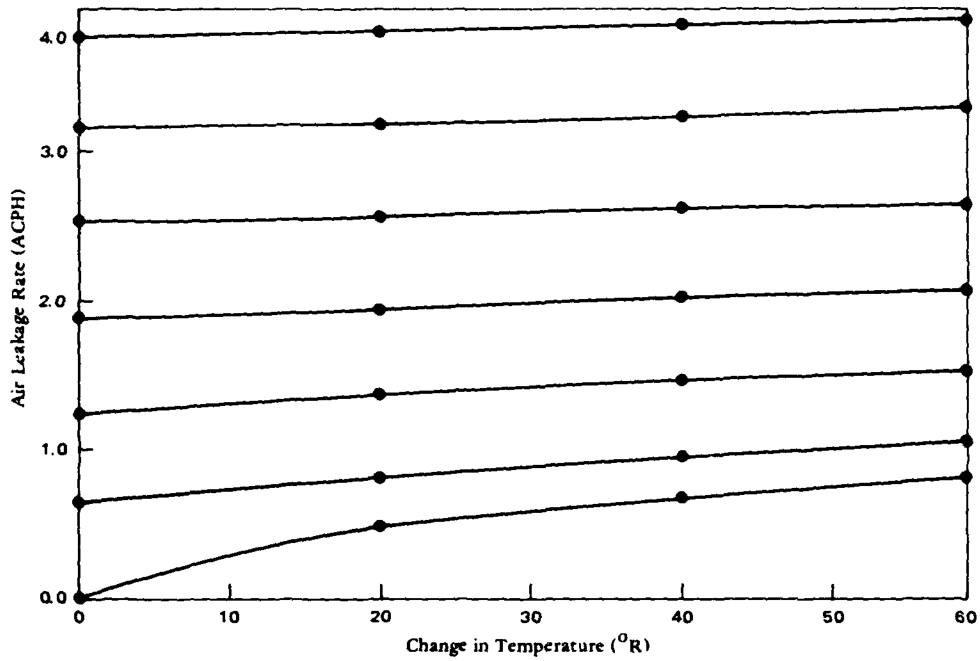


Figure 6. Air infiltration rate versus temperature for Hangar V-88, NARF Norfolk.

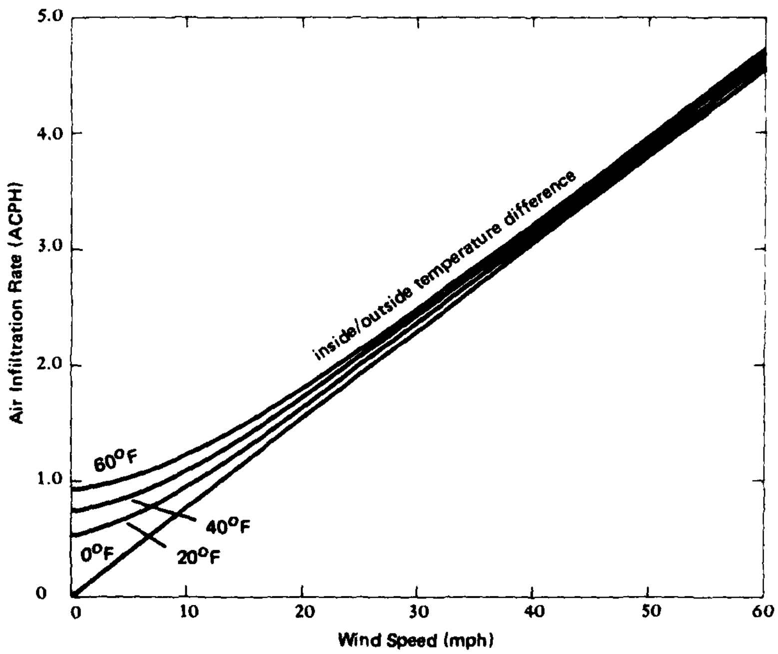


Figure 7. Air infiltration rate versus wind speed (door closed) for Hangar 365, McClellan AFB.

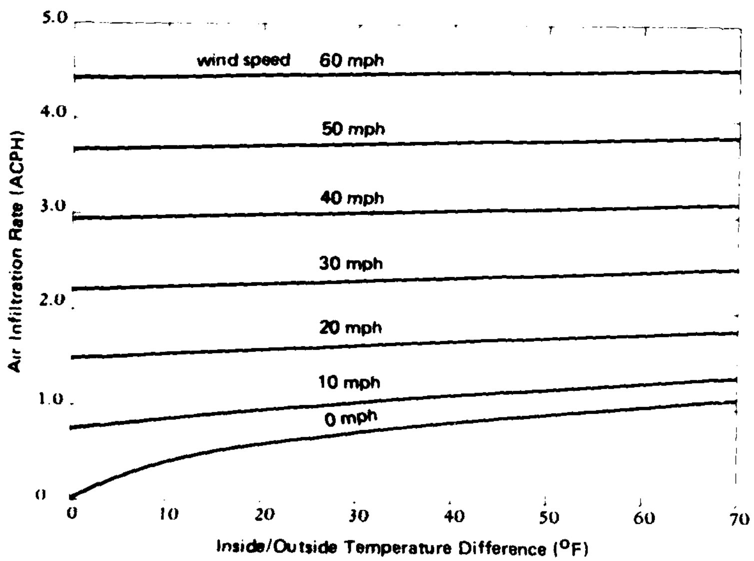


Figure 8. Air infiltration rate versus temperature differential (door closed) for Hangar 365, McClellan AFB.

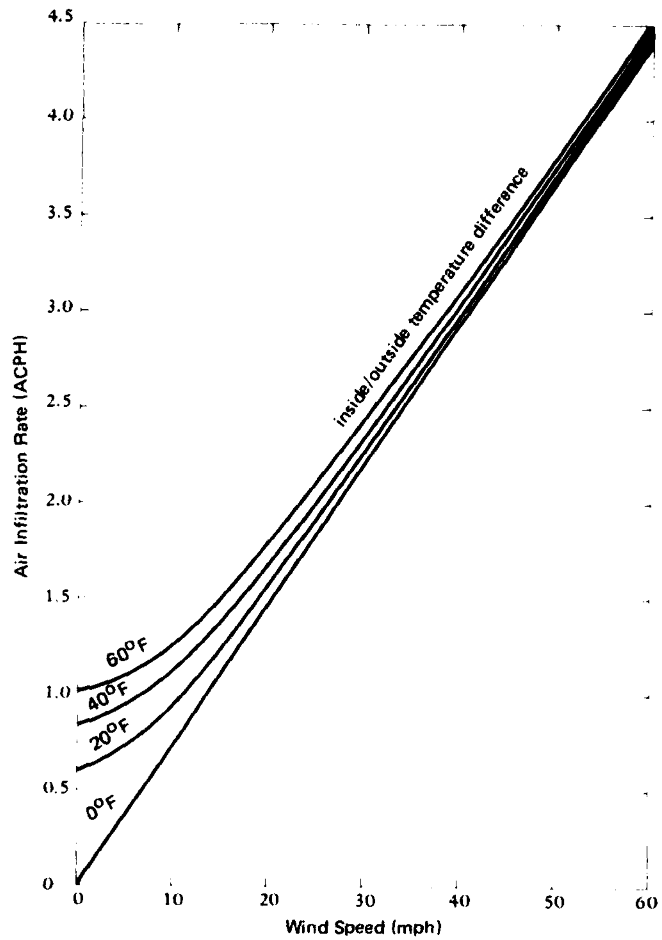


Figure 9. Air infiltration rate versus wind speed (door closed) for Hangar 250, NAS Brunswick.

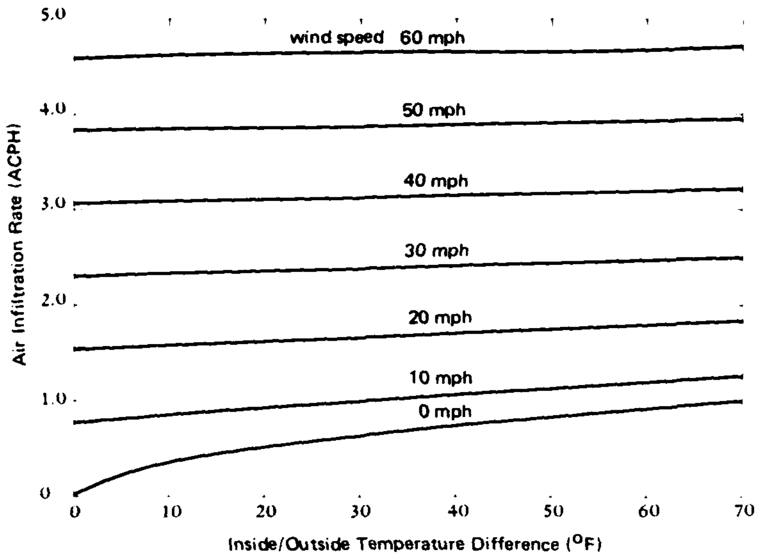


Figure 10. Air infiltration rate versus temperature differential (door closed) for Hangar 250, NAS Brunswick.

Table 3. Hangar Crack and Vertical Surface Areas Measured at Four Activities

Building No.	Activity	Vertical Surface Area (ft^2)	Total Crack Area (ft^2)	% Crack Area to Vertical Surface Area
V-147	NARF Norfolk	61,000	151	0.25
V-99	NARF Norfolk	46,000	106	0.23
250	NAS Brunswick	33,000	75	0.23
867	Minot AFB	24,000	79	0.34
365	McClellan AFB	23,000	53	0.23

Equation 9 can be expressed as:

$$Q = \frac{0.23A_w}{100}\sqrt{79.34H^{1/2}\ S^3 + 32.43\Delta T} \tag{10}$$

or

$$I = \frac{0.138A_w}{V}\sqrt{79.34H^{1/2}\ S^2 + 32.43\Delta T} \tag{11}$$

Hangar air infiltration rate design tables have been developed from Equation 11 and are included as Appendix A to this report for the following conditions:

- Hangar volumes from 500,00 to 4,000,000 ft^3
- Hangar heights from 35 to 50 feet
- Design wind speeds from 0 to 40 mph
- Outside design temperature from -10 to 30°F

Energy loss due to air infiltration can be calculated by Equation 12:

$$e = \rho \, c_p \, \Delta T \, Q/\eta = \frac{\rho \, c_p \, \Delta T \, A_o}{\eta} \sqrt{79.34H^{1/2} \, S^2 + 32.43\Delta T} \qquad \text{. . . . . (12)}$$

where: e = hourly loss, Btu/hr

ρ = density of air, lb/ft³

c_p = specific heat for air, Btu/lb-°F

η = overall heating system efficiency, %/100

An annual air infiltration energy loss can be estimated by calculating and summing loss for each hour of a heating season or by using degree-days, average heating season wind speed, and values of ρ and c_p at 60°F. A degree-day, d, is generally expressed as:

$$d = 65 - T_A \qquad (13)$$

where T_A is the average outside temperature for the day. The total number of degree-days during the heating season, D, can be expressed as

$$D = \sum_{i=1}^{N} d = \sum_{i=1}^{N} (65 - T_A) \quad \text{for } T_A < 65 \qquad (14)$$

where N is the total number of days in a heating season.

The average inside/outside temperature difference for a hangar for a heating season, ΔT_A, may then be expressed as:

$$\Delta T_A = \frac{D}{N} \qquad (15)$$

A rough approximation of air infiltration energy losses during a heating season, L_A, can be made by assuming that a heating season has 150 days with an average wind speed of 10 mph and by substituting Equation 15 into Equation 12; thus

$$L_A = \frac{\rho \, c_p \, D \, A_o}{\eta} \sqrt{7{,}934 \, H^{1/2} + 0.2162D} \qquad (16)$$

During the air infiltration measurements, inspections were conducted to identify common air leakage points. The three typical sources are doors, surface defects, and windows. Typical defects for each source are listed as follows:

Source	Defect
Doors	seals misuse misalignment faulty closure mechanisms
Surface Defects (walls, roof, and doors)	gaps at wall, roof, and floor interface holes cracks storm drains penetrations, piping, and electrical systems
Windows	seals broken or missing panes caulking warped frames inoperative closure mechanisms

These sources can be grouped into three categories:

- maintenance items
- design
- operational

By estimating the total crack area associated with these sources and using Equation 16, the annual energy losses caused by air infiltration in a hangar can be estimated.

AIR STRATIFICATION

Measurements of interior air temperature gradients (stratification) were made in Air Force and Navy hangars at various locations in the United States as part of the investigation on methods to reduce the energy required to heat hangars. These measurements indicated that stratification is a typical condition in heated hangars. In addition, the data indicated that the magnitude of stratification is not a function of hangar height but a function of the temperature difference between the outside (ambient) air and inside floor-level air temperatures. Measurement of hangar stratification is difficult because of the height and inaccessibility of ceiling areas; however, these data are required to determine the potential magnitude of energy saved by adding roof insulation or by installing destratification systems.

Table 4 presents the floor, ceiling, and ambient temperatures measured at various heights in the hangars. The measurements were divided into groups by floor-level air temperature (58, 65, and 72°F). Plots were made of ceiling-level air temperatures versus ambient air temperature for each floor-level temperature group and a curve fit analysis performed. The results of these plots and analyses are presented in Figure 11. Temperature gradients were normalized and averaged. A plot of an average hangar floor to ceiling temperature gradient is presented in Figure 12.

Table 4. Hangar Stratification Temperature Measurements With Various Ceiling Heights

Air Temperature (°F)			Ceiling Height (ft)
Floor Level	Ceiling Level	Ambient	
50	63	34	45
57	71	37	60
58	76	31	60
59	75	34	45
64	76	42	52
64	73	45	35
65	78	43	60
65	81	34	60
72	86	34	55
73	100	31	60

The following exponential curve was common for all hangar ceiling level temperatures:

$$T_c = \beta e^{-\gamma T_a} \tag{17}$$

where: T_c = inside air temperature 1 ft below the ceiling, °F

T_a = ambient air temperature, °F

β = $f(T_f)$

γ = $f(T_f)$

T_f = inside air temperature 1 foot above the floor, °F

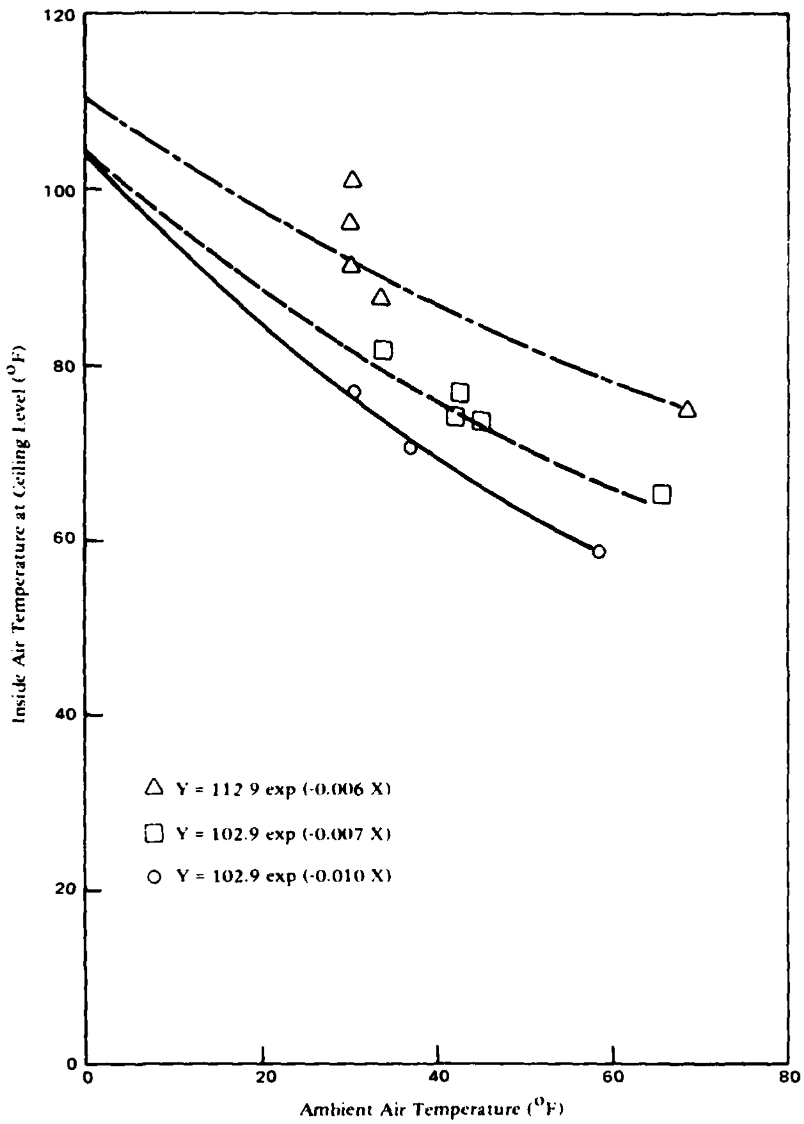


Figure 11. Exponential curve fit, with floor level air temperature at 58, 65, and 72°F.

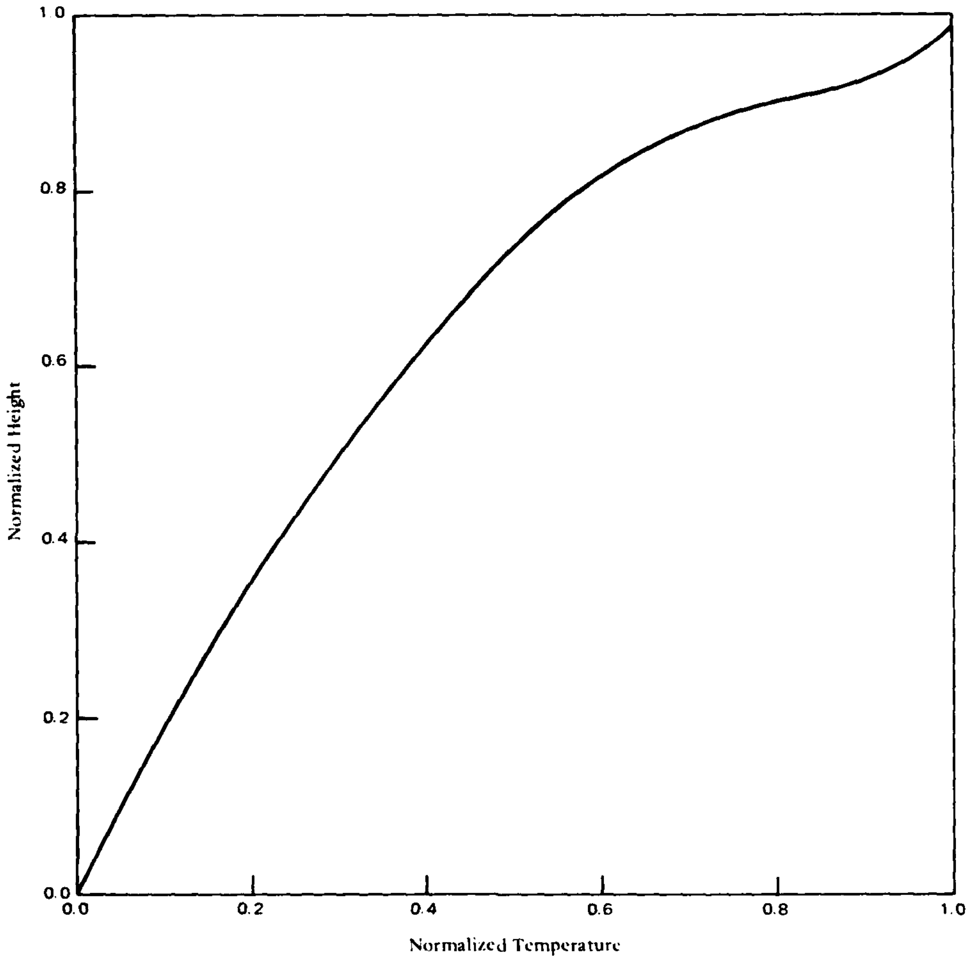


Figure 12. Hangar temperature gradient profile.

Figures 13 and 14 were plotted for β and γ versus floor-level air temperature and curve fit analyses performed. The optimum values of β and γ were determined and are presented as Equations 18 and 19.

$$\beta = 63.12 + 0.664\ T_f \qquad (18)$$

$$\gamma = 40.8 T_f^{-2.065} \qquad (19)$$

By substituting Equations 18 and 19 for β and γ, Equation 17 can be expressed as:

$$T_c = (63.12 + 0.664\ T_f)\ e^{(-40.8 T_f^{-2.065}) T_a} \qquad (20)$$

Equation 20 can be used to estimate the ceiling-level air temperature in buildings such as hangars, where no significant industrial heat sources exist. The accuracy of the equation is indicated by Table 5. Figure 15 presents plots of Equation 20 for floor-level air temperatures of 50, 60, 70, and 80°F versus ambient air temperature.

Two general observations can be made concerning stratification in heated high-ceiling structures such as hangars:

1. Stratification is predominantly a function of inside floor-level air temperatures and outside air temperatures.

2. The severity of stratification can be reduced by maintaining the inside air temperature as low as possible with consideration of the structure's utilization.

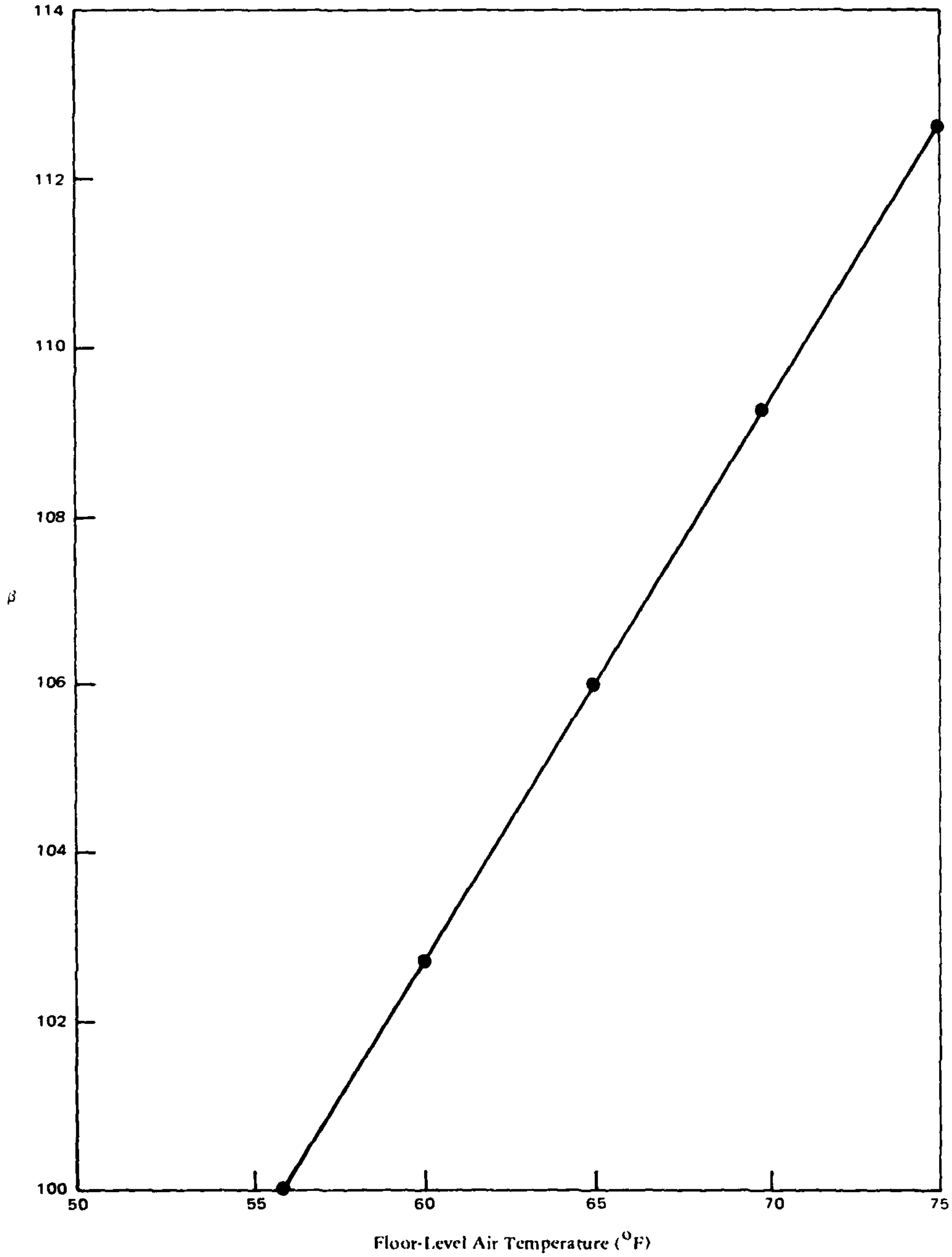


Figure 13. β versus floor-level air temperature.

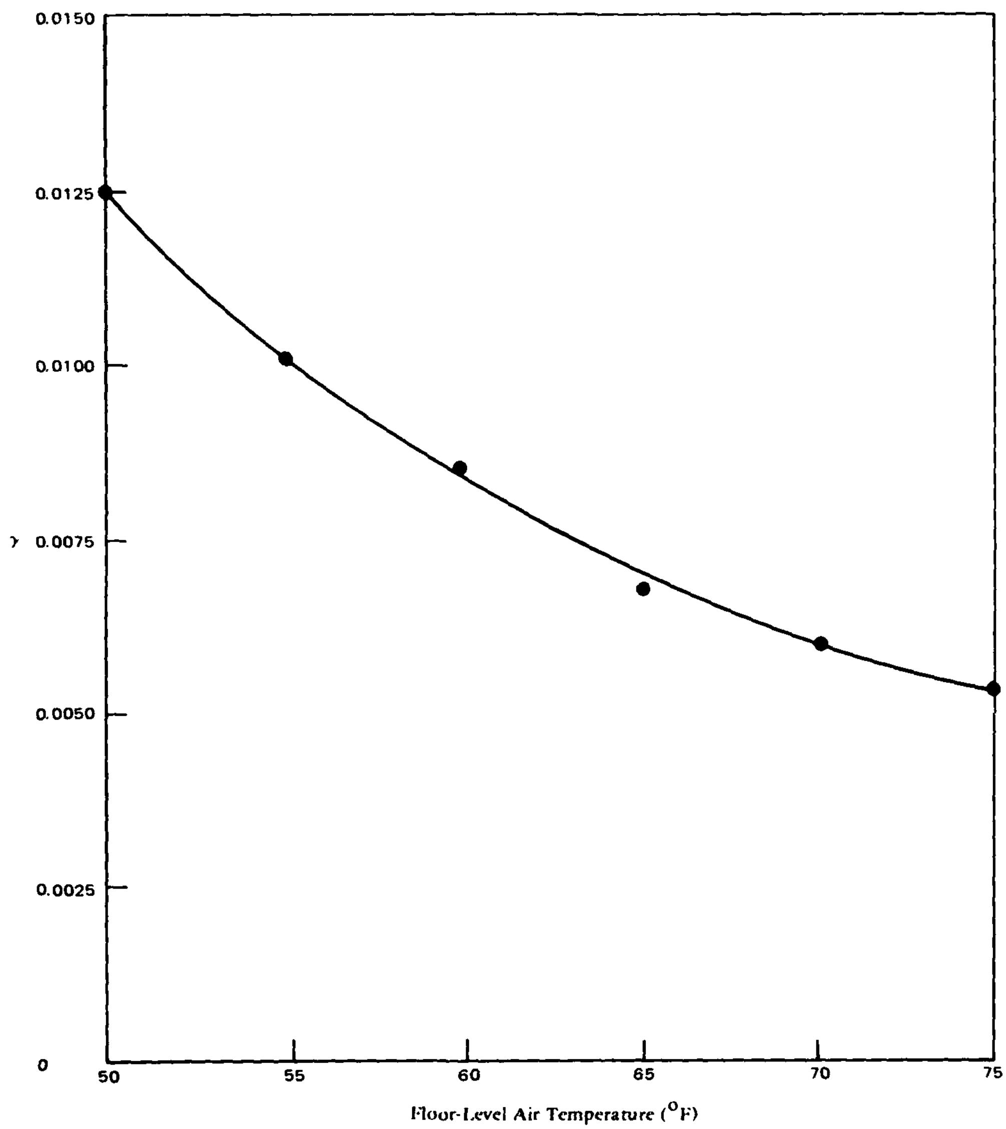


Figure 14. γ versus floor-level air temperature.

Table 5. Comparison of Calculated and Measured Ceiling Level Temperatures

Air Temperature (°F)				Error (%)
Floor Level	Ambient	Ceiling Level		
		Measured	Calculated	
50	34	63	63	0
57	37	71	71	0
58	31	76	76	0
59	34	76	75	2
64	42	76	77	3
64	45	73	75	7
65	43	78	77	3
65	34	81	83	4
72	34	86	91	10
73	31	100	93	10

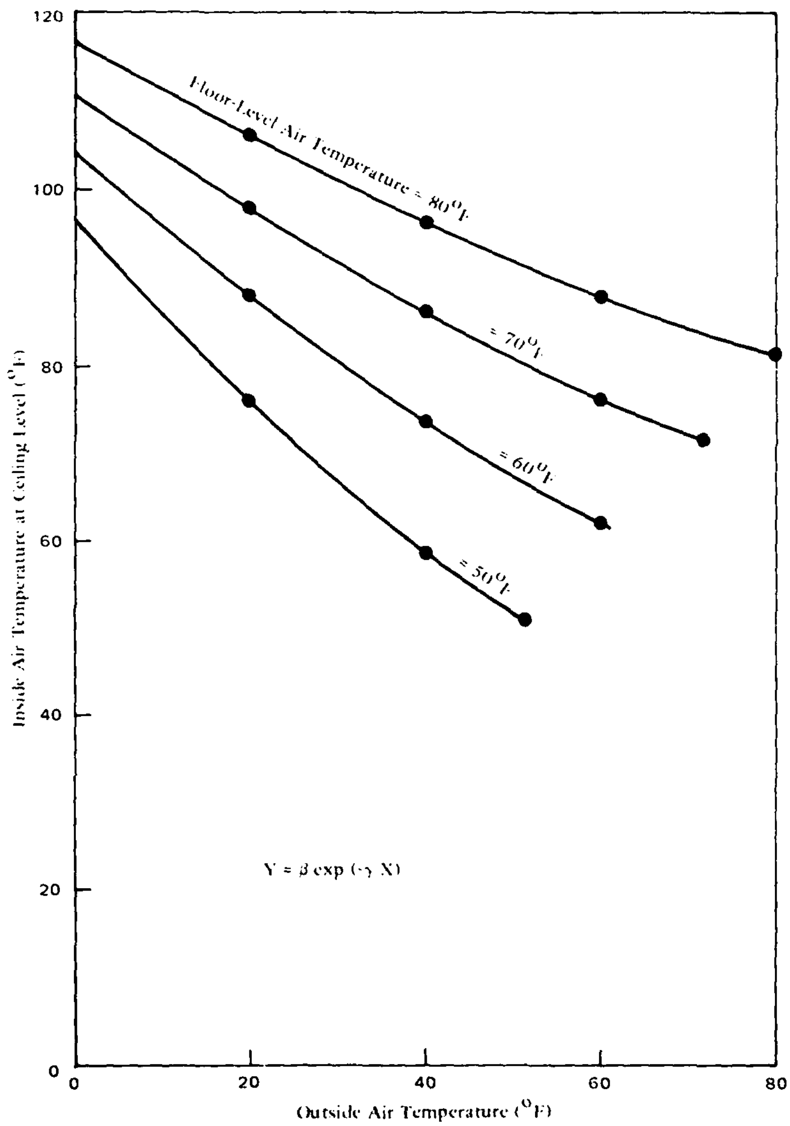


Figure 15. Ceiling-level air temperature characteristics.

POSSIBLE SOLUTIONS TO ENERGY LOSS PROBLEMS

Several solutions have been proposed and are being used. These are discussed in this portion of the report.

NYLON BRUSH SEALS

The Patuxent River Naval Air Station has installed nylon brush seals on hangar aircraft access doors. The nylon seals were easily installed by Public Works personnel and have none of the maintenance problems, such as cracking and deformation, normally associated with the rubber seals currently used on most military installations. While the nylon brush seal was cost-effective based upon reduction in seal replacement costs, data were not available on its effects upon hangar energy consumption.

Controlled tests were conducted at NCEL to measure the difference, if any, in air infiltration rates with rubber seals and with nylon brush seals. These items were used to seal an opening in a pressure chamber; a variable speed blower, calibrated for air flow versus the pressure difference across the blower, was used to pressurize the chamber. Table 6 and Figures 16 and 17 present the test results. Curve fit analyses were used to define the following empirical equations associated with the test results:

For rubber seals,

$$Q = 782S^{1.0574} \text{ per 100 ft of seal, ft/min}^3 \qquad (21)$$

For nylon brush seals,

$$Q = 521S^{1.0157} \text{ per 100 ft of seal, ft/min}^3 \qquad (22)$$

Air leakage rates using rubber and nylon brush seals were calculated using Equations 21 and 22 for wind speeds ranging from 1 to 20 mph, and a curve fit analysis was used to develop the following empirical equation for the differential leakage between the seals:

$$\Delta Q = 266\ \bar{S}^{1.1215} \text{ per 100 ft of seal, ft/min}^3 \qquad (23)$$

where $\bar{S}$ is the average wind speed during the heating season in miles per hour.

Table 6. Results of NCEL Leakage Tests

Wind Speed (mph)	Leakage[a] With--	
	Rubber Seal (cfm/100ft)	Brush Seal (cfm/100 ft)
5.1		3,000
5.1		2,500
5.2	4,800	
5.4	5,000	
6.3	5,100	3,200
6.6	6,000	3,500
7.0		4,000
7.5	6,200	
8.0	7,300	4,200
10.0	7,500	5,600
10.5	9,100	5,900
12.2		6,400
12.3	10.700	
12.5		7,200
13.0	12,100	
14.4		7,500
14.8	14,000	
15.5	15,100	8,500
16.6		9,000
17.2	15,700	
17.5		9,100
17.7		10,000
18.0	17,100	
18.2	16,400	
18.7		9,800
18.9		11,000
19.2	18,300	
19.5	17,800	
20.0		11,300
20.6	19,800	

[a]Blanks indicate data not taken.

Annual air leakage may then be written:

$$\Delta Q_y = 24 \times 60 \times 266 \, N\bar{S}^{1.1215} \text{ per 100 ft of seal}$$

where ΔQ_y is the differential air leakage over an entire heating season (ft^3/yr).

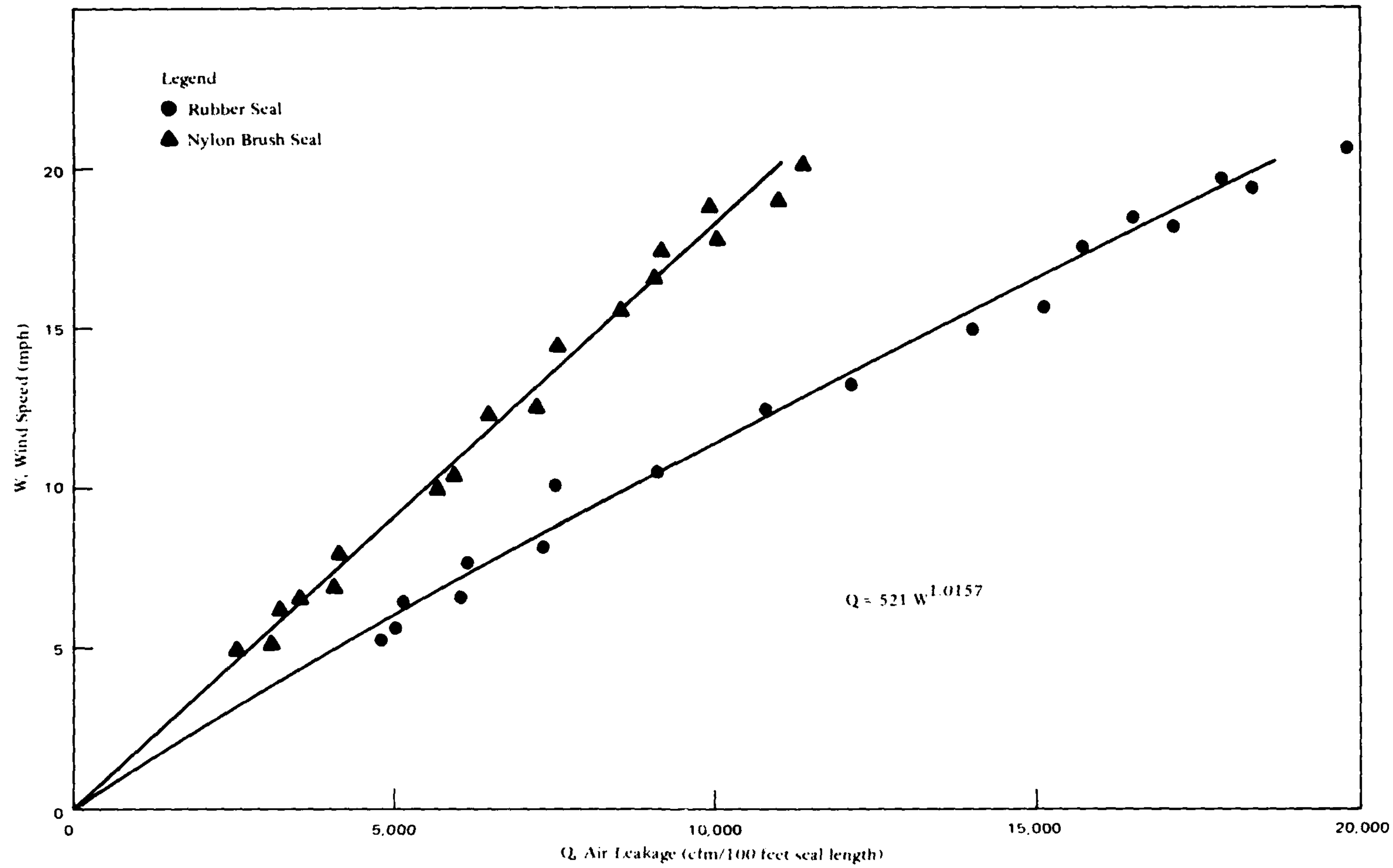


Figure 16. Air leakage tests with nylon brush seals and rubber seals.

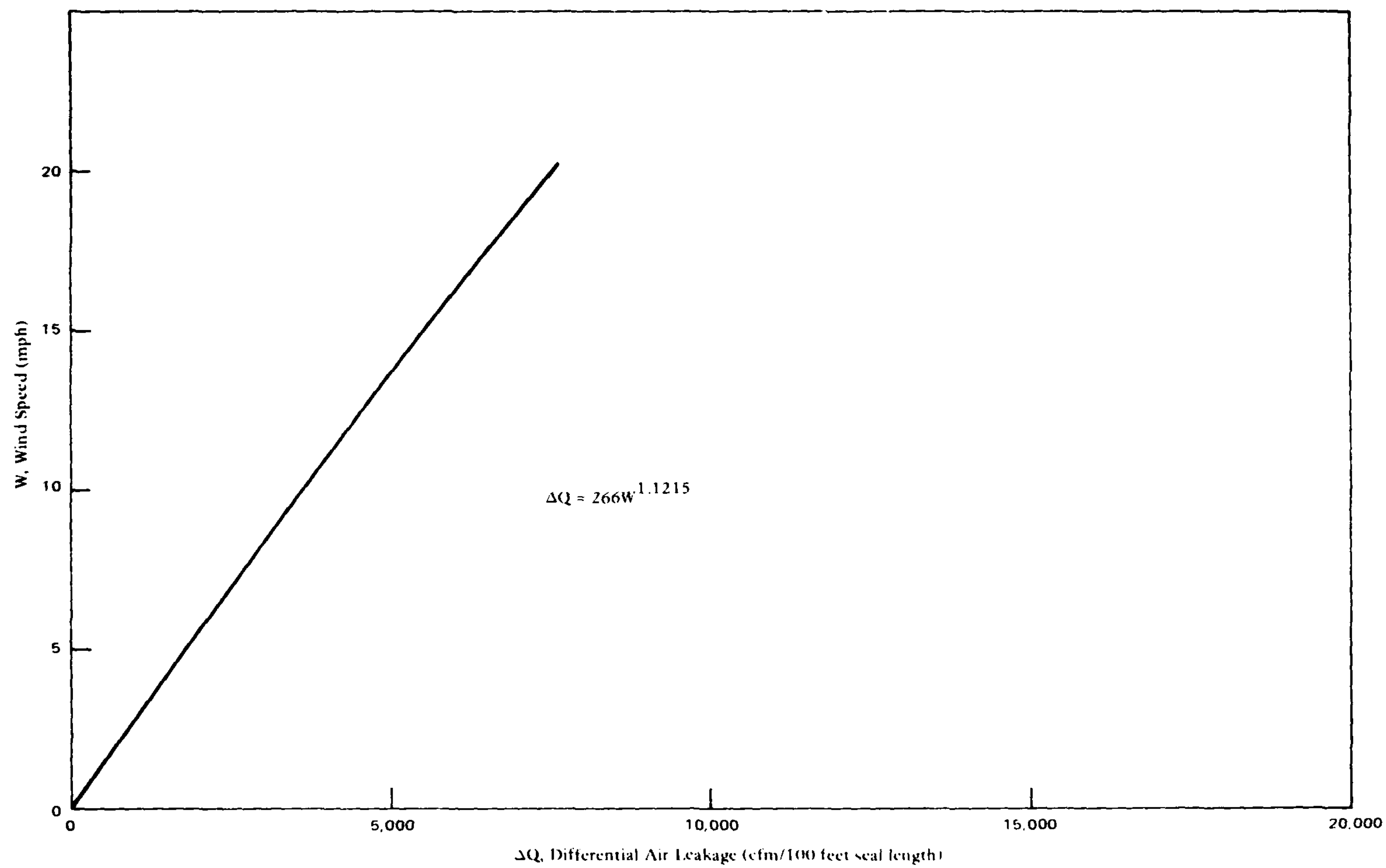


Figure 17. Differential curve resulting from air leakage tests of rubber seals and nylon brush seals.

By approximating the average annual heating season inside/outside air temperature difference by Equation 15, $\rho = 0.076$ lb/ft³, and $c_p = 0.24$ Btu/lb-°F, the annual energy loss, L_A, can be estimated from:

$$L_A = \rho c_p \Delta T_A \Delta Q_y$$

or

$$L_A = \frac{0.007D}{\eta} \bar{S}^{1.1215} \text{ MBtu per 100 ft of seal} \quad (24)$$

Figure 18 presents the annual energy reduction obtained by using nylon brush seals versus average heating season wind speed for overall heating system efficiency of 60, 70, 80, and 100%.

VINYL STRIP DOORS

Vehicle and personnel movement in and out of hangars is an air infiltration source. Vinyl strip doors (Figure 19) are available to reduce this air leakage source; however, no data have been available to document their effectiveness. Tests were conducted at NCEL to measure the residual air leakage through vinyl strip doors versus wind speed. The results of the tests are shown in Figure 20.

Equations 25 and 26 are empirical equations developed by using curve fit analyses for the data presented in Figure 20.

$$Q_o = 1000\, A_d \sqrt{\Delta P} \quad (25)$$

$$Q_v = 2300\, A_d\, (\Delta P)^{0.86} \quad (26)$$

where: A_d = door area, ft²

ΔP = hangar's inside/outside pressure difference, psi

Q_o = air leakage, open door, lb/min

Q_v = air leakage, vinyl strip door; lb/min

The reduction in air leakage, Q_r, obtained from fitting doors and other large openings with vinyl strip doors can then be calculated from:

$$Q_r = Q_o - Q_v, \text{ lb/min} \quad (27)$$

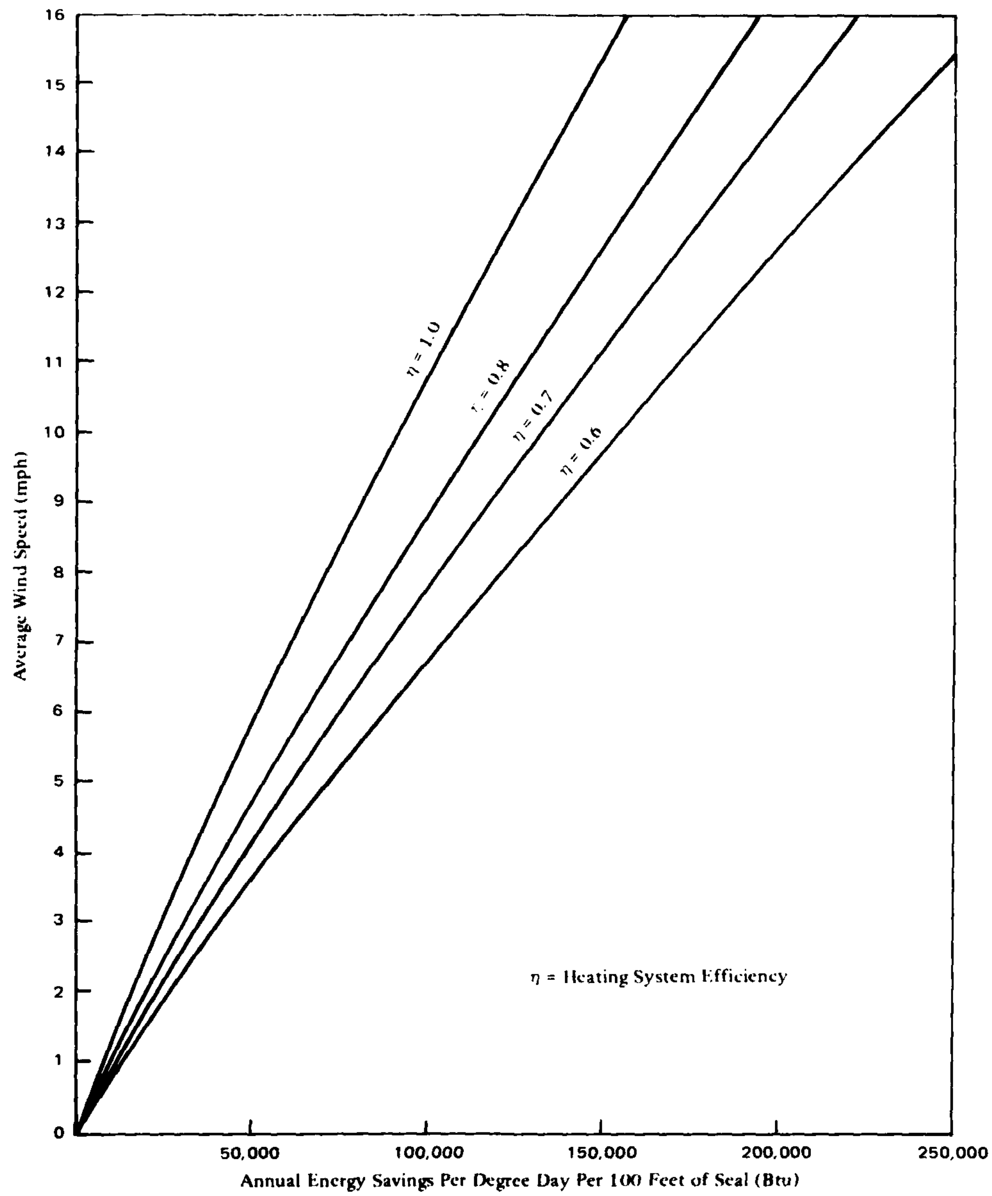


Figure 18. Annual energy savings with nylon brush seals.

Figure 19. Flexible vinyl strip doors.

The inside/outside pressure difference of a hangar is a function of wind velocity (speed and direction), inside/outside air temperature difference, building configuration, and building tightness. The temperature difference effect is negligible for most structures when the wind speed exceeds 5 mph (Ref 1).

For wind speeds below 30 mph, the inside/outside pressure differences* for hangars, industrial shops, and similar structures may be approximated in pounds per square inch by (Ref 5):

$$\Delta P = 9 \times 10^{-6} S^2 \tag{28}$$

Energy Savings

The energy savings, E_V, obtained by installing flexible vinyl strip doors is a function of the reduced rate of air leakage, the length of time a door remains open, the inside/outside temperature difference and the heating system's overall efficiency. It may be calculated from:

$$E_V = 60 \; c_p \; t \; Q_r \; \Delta T/\eta \tag{29}$$

where: E_V = energy saved, Btu

t = open door duration, hr

Design Considerations

Flexible vinyl strip doors can be either header- or wall-mounted (Figures 21 and 22 and Table 7). Header or lintel mounts bolt the inside door frames to the door header or lintel. Wall mounts bolt to the interior wall and are the preferred mounting because the possibility of vehicles striking and damaging the mounts is reduced. The wall mounting strip should be at least 2 inches above the door header and extend to at least 6 inches on either side of the door opening. Header mounts require an 18-inch clearance between the lintel and the highest point of a vehicle passing underneath.

Two mounting patterns are possible: lap and shiplap. Lap patterns (Figure 23) are for conditions where ease of opening the vinyl door is the primary concern. Shiplap patterns (Figure 24) are for exterior openings or for environmental isolation (to control wind, noise, cold, or other condition).

*Average pressure difference for wind direction of 0, 45, 90, 135 and 180 degrees relative to the structure.

Table 7. Design Guidelines for Vinyl Strip Doors

Application	Typical Strip Specifications (in.)		
	Thickness	Width	Overlap
Lightweight foot traffic closures only	0.060	6	3[a]
Freezers, coolers, and refrigerated trucks	0.080	8	2[b]
Interior doors to 8 ft high	0.080	8	4[a]
Interior/exterior openings from 8 to 14 ft high	0.120	12	6[a]
Exterior doors, heavy vehicular traffic	0.160	16	8[a]

[a]Full lap (see Figures 24 and 25).

[b]Two-third lap (see Figures 24 and 25).

With flexible vinyl strip doors for hangar and similar applications, five areas of concern are:

1. Vinyl, a flammable material, produces dense smoke when burning; only flame-resistant vinyl doors should be installed.

2. Vinyl strip doors can undergo violent motion in high winds and present a safety hazard to personnel.

3. In cold dry weather, static electricity can accumulate on vehicles passing through the doors; vehicle grounding strips may be required.

4. Vinyl, while a relatively strong material, creeps under load; applications to door openings with heights in excess of 16 feet should be carefully coordinated with the manufacturer of the vinyl strip door.

5. The bottom edge and other surfaces of the flexible doors crack and pieces chip off the vinyl strips, which could cause aircraft damage unless caution is observed.

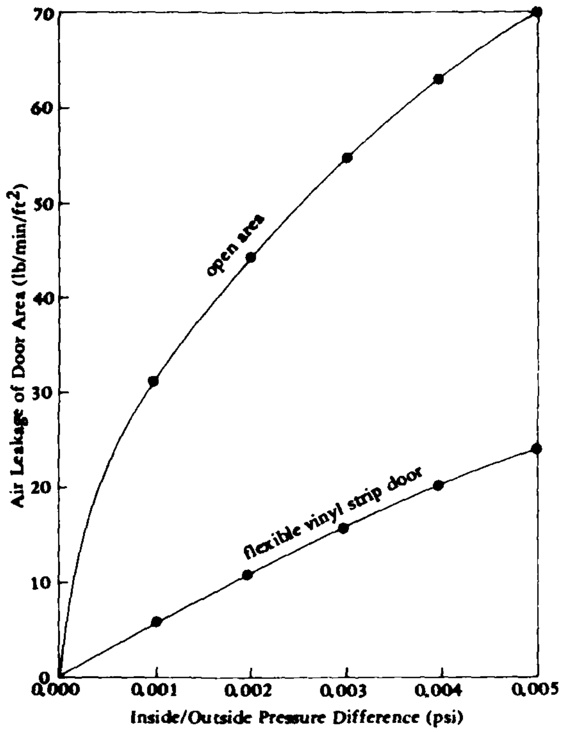


Figure 20. Air leakage characteristics of flexible vinyl strip doors.

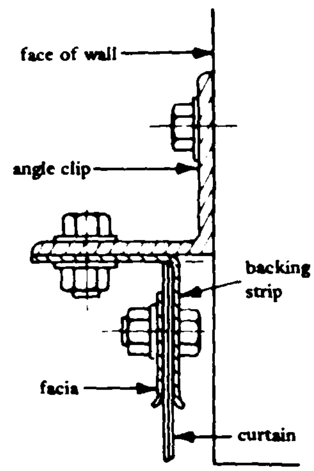


Figure 21. Lintel mounting method.

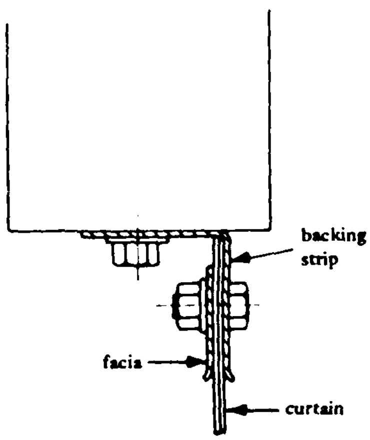


Figure 22. Wall mounting method.

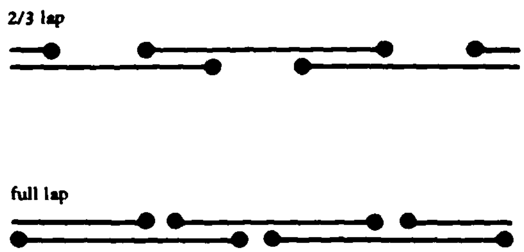


Figure 23. Lap pattern for conditions where ease of opening strip is primary concern.

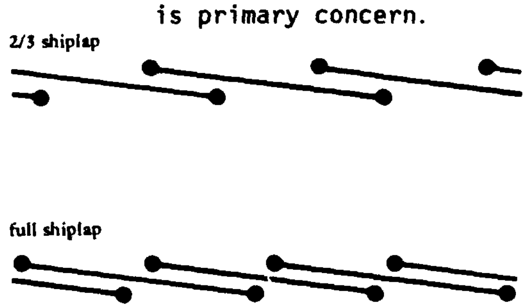


Figure 24. Shiplap pattern for exterior opening (facing outside); *for conditions of wind, draft, noise, and cold.*

DESTRATIFICATION

Measurements made in five hangars (two Air Force and three Navy) indicated that stratification is a typical phenomenon in heated hangars (Ref 1). This phenomenon results in increased energy consumption because of the following:

- temperature difference across the roof and upper wall surfaces increases the amount of heat transferred from inside a structure to the outside
- the chimney effect increases the structure's air infiltration rate
- unused heat is wasted

Five destratification concepts (three commercial, one developed by NCEL and one suggested by Atlantic Division* of the Naval Facilities Engineering Command) were evaluated to determine the effectiveness and adaptability of each concept to hangar application. The results of the evaluations of all the concepts are presented in this section of the report, as well as the design criteria for the NCEL destratification concept, which is recommended for use.

Concepts

The five destratification concepts evaluated are described as follows:

- Destratifier tube - commercial (Figure 25): The unit consists of a small blower mounted on top of a tube or duct which goes from floor to ceiling . The fan blows hot ceiling-level air down to the floor level where the hot air mixes with cooler floor-level air.
- Ceiling fan - commercial (Figure 26): A fan, mounted at the ceiling level, blows hot air down toward the floor where it is mixed with the cooler floor-level air.
- Floor blower - commercial (Figure 27): A blower placed at the floor level blows cool floor-level air upward through a duct toward the ceiling where it mixes with the hot ceiling-level air.

*LANTDIV.

Figure 25. Destratifier tube.

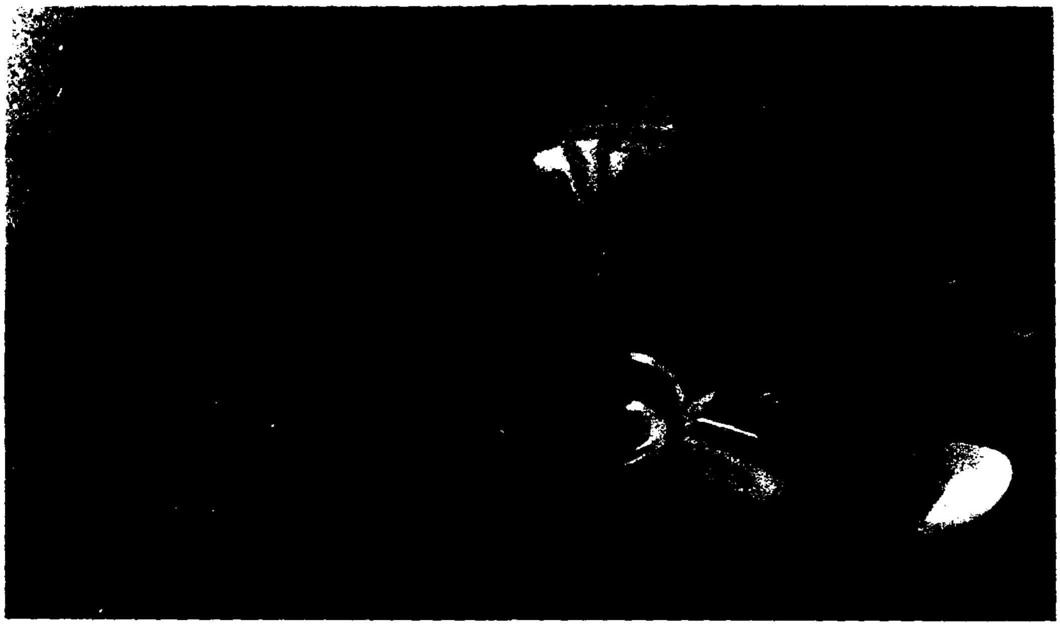


Figure 26. Ceiling fan.

- Cold air jet - NCEL (Figure 28): A blower sucks cool floor-level air through a duct and injects this air as a high velocity air jet in the ceiling where it mixes with the hot ceiling-level air.

- Heating system modification - LANTDIV (Figure 29): Hot ceiling-level air is used as the intake air for the heating system's heating coils. The intake air can be routed to the heating system via a duct, or the heating coils' air intake can be located within the hot ceiling-level air.

Preliminary Evaluation

The three commercial destratification units were evaluated at NCEL to determine their adaptability for use in hangars. All three were installed in a shop building at the Laboratory and their effectiveness were measured. The results of the evaluation are provided in Tables 8, 9, and 10 for the destratifier tube, ceiling fan and cold air blower, respectively. Figure 30 shows where the data were taken and the location of the destratifier within the building. As can be readily noted from the tables, while neither the ceiling fan nor the destratifier tube produced any significant changes in the building's stratification characteristics, the cold air blower rapidly destratified the building.

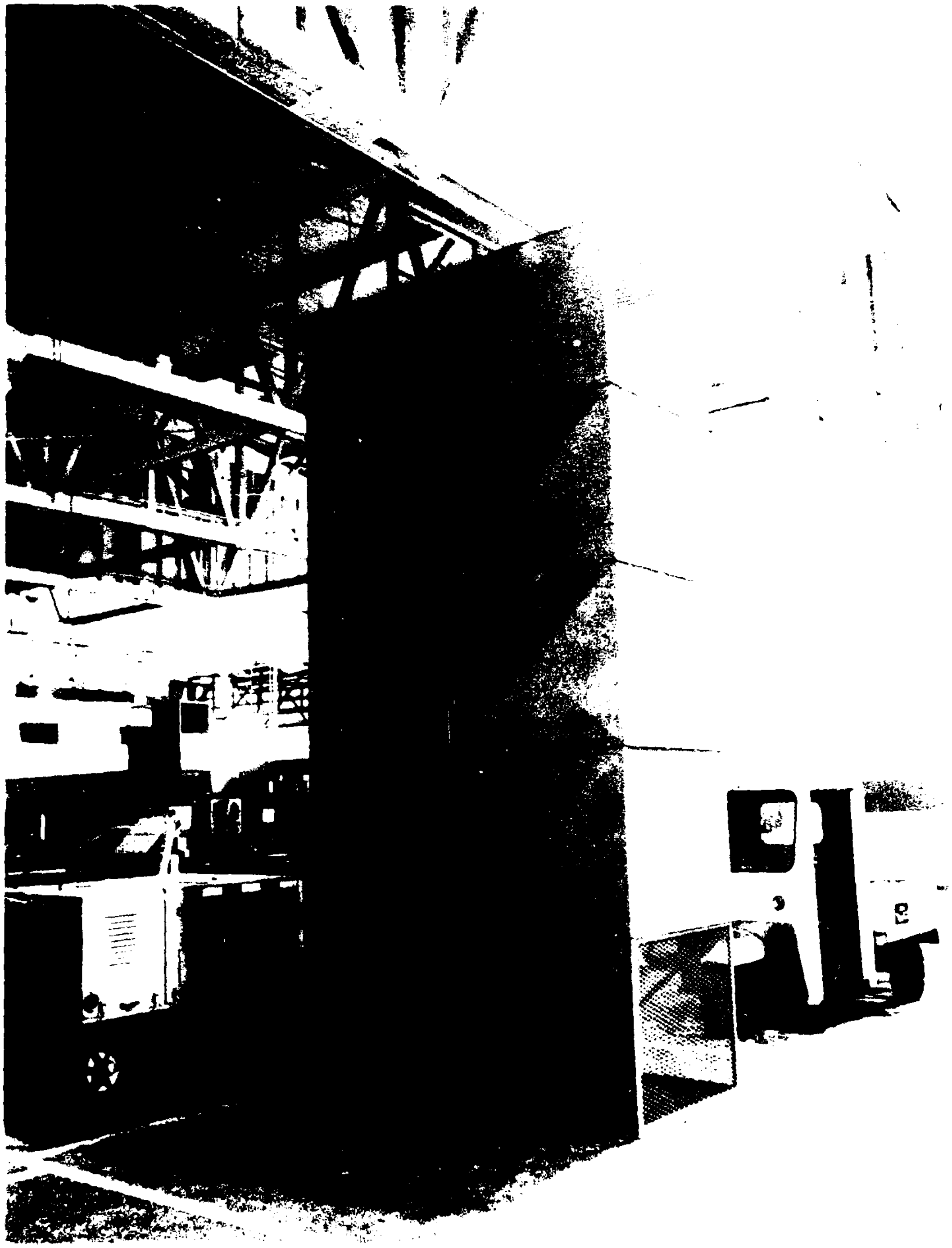


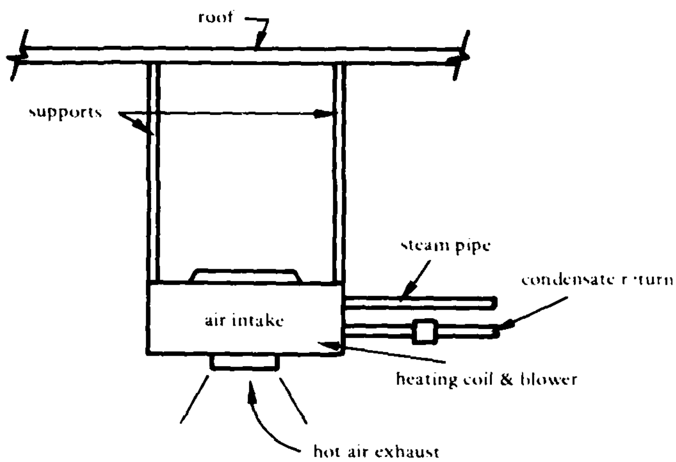
Figure 27. Cold air blower.

(a) Air intake.

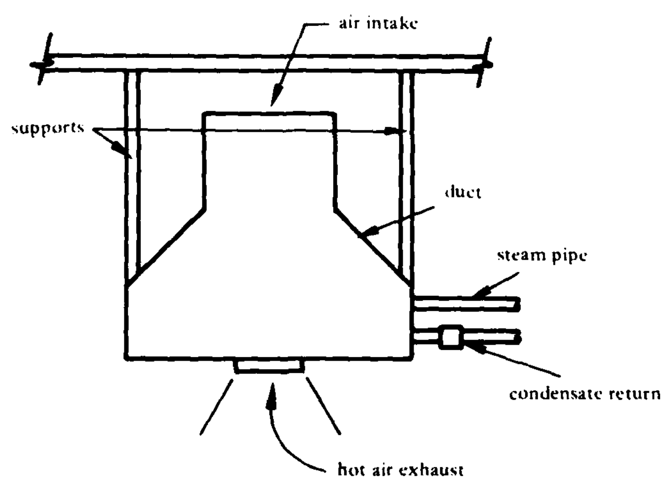
(b) Blower.

(c) Blower and duct.

Figure 28. NCEL cold air jet.



(a) Existing configuration.



(b) LANTDIV modification.

Figure 29. LANTDIV heating system modification.

Table 8. Destratifier Tube Evaluation Building 564, 4 February 1982

[Weather, cloudy]

Time	Data Point[a]	Temperature (°F) at Following Test Locations				Average Temperature[b] (°F)
		1	2	3	4	
(a) Before Tube Operation Begun; Ambient Temperature, 60°F						
0830	Floor	68	69	68	68	68
0838	Loft	73	74	72	74	73
0847	Ceiling	95	94	89	92	93
(b) After Tube Operation Begun at 0917, Ambient Temperature, 62°F						
1240	Floor	67	68	69	68	68
	Loft[c]	-	-	-	-	-
1247	Ceiling	95	94	93	94	94

[a]See Figure 31(a) for data points and test locations.

[b]Floor/ceiling temperature difference: (a) 25°F; (b) 26°F.

[c]Data not taken at this level.

Table 9. Ceiling Fan Evaluation Building 564, 4 February 1982

Time	Data Point[b]	Temperature (°F) at Following Test Locations				Average Temperature[b] (°F)
		1	2	3	4	
(a) Before Fan Operation Begun						
1240	Floor	67	68	69	68	68
	Loft	-	-	-	-	-
	Ceiling	95	94	93	94	94
(b) After Fan Operation Begun at 1300; Ambient Temperature, 63°F; Weather, Cloudy						
1330	Floor	73	72	74	73	73
	Loft	77	77	76	76	77
	Ceiling	97	95	93	89	94

[a]See Figure 31(b) for data points and test locations.

[b]Floor/ceiling temperature difference: (a) 26°F; (b) 21°F.

Table 10. Floor Air Blower Test Results, Building 564

[Weather, rain; ambient temperature, 53°F]

Time	Data Point[a]	Temperature (°F) at Following Test Numbers					Average Temperature[b] (°F)
		1	2	3	4	5	
(a) Before Air Blower Operation Begun							
1000	Floor	76	76	76	75	74	75
1005	Loft high	77	81	78	77	74	77
1010	Ceiling	88	94	94	84	74	87
(b) After Air Blower Operation[c] Begun at 1015							
1025	Floor	77	77	79	76	77	77
1030	Loft high	75	76	80	75	77	77
1035	Ceiling	76	76	84	75	80	78

[a]See Figure 31(c) for data points and test locations.

[b]Average floor/ceiling temperature difference: (a) 12°F; (b) 1°F.

[c]Electric power consumption: 850 watts.

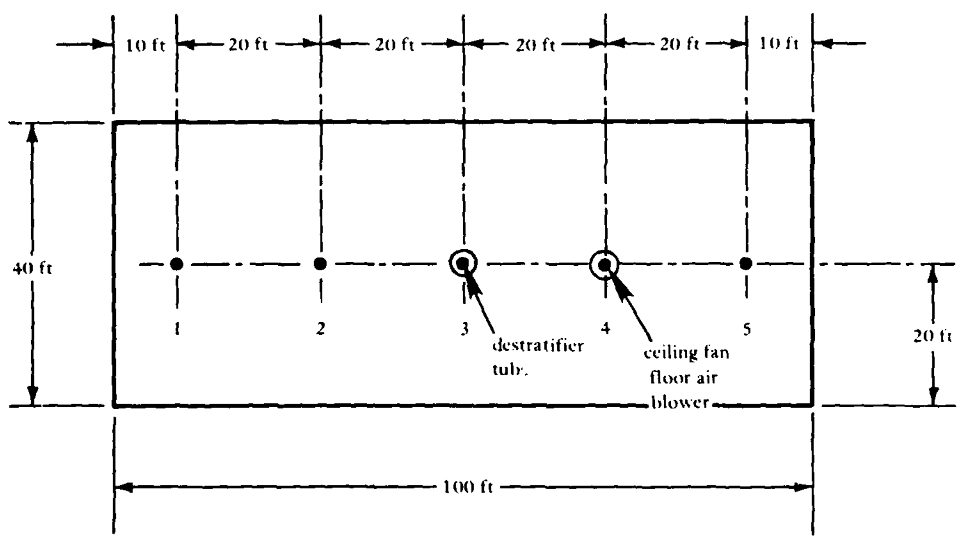


Figure 30. Locations of data points and destratifiers.

The destratifier tube and ceiling fan were installed in a test chamber (12 feet wide x 12 feet long x 14 feet high) in order to determine if their effectiveness and installation criteria could be established. Tables 11 and 12 and Figures 31 and 32 provide the results of the chamber evaulation for the destratification tube and the ceiling fan, respectively. One installation parameter not determined was a maximum ceiling height. Although all three commercial units can destratify a structure - to varying degrees - and can save energy if properly installed, only the cold air blower indicated that it might be practicable for destratifying spaces with heights typical of a hangar. The number of units, height, and destratification effectiveness would have to be determined.

Table 11. NCEL Destratifier Tube Test Results

Item	Measurement
ΔT_S (floor/ceiling temperature difference, stratified)	38°F
ΔT_D (floor/ceiling temperature difference, destratified)	31°F
Destratification efficiency, $\dfrac{(\Delta T_S - \Delta T_D) \times 100}{\Delta T_S}$	18.5%
Test chamber volume[a], V	2,500 ft³
Destratifier fan air movement, Q_f	6,300 ft³/hr
Destratifier flow-to-volume ratio, Q_f/V	2.5
Destratifier electric power consumption, p	100 watts
Number of destratifier units required for installation in a building	$\dfrac{\text{Volume of space}}{2.5 \times Q_f}$

[a]Test chamber height = 14 feet.

Table 12. NCEL Test Results, Ceiling Fan

Item	Measurements Taken at Points--				
	1	2	3	4	5
Fan speed, rpm	240	160	120	90	60
Fan flows, q_f, ft^3/min	1,990	1,328	996	747	498
Fan flow, Q_f, ft^3/hr	119,000	79,680	59,760	44,820	29,880
ΔT_S, °F	32	29	29	29	32
ΔT_D, °F	6	10	17	27	32
Destratification efficiency, %	81	66	41	1	0
Test chamber volume, V^a, ft^3	2,500	2,500	2,500	2,500	2,500
Q_f/V	47.5	32	24	18	12
Power, P, watts	173	150	138	127	115

[a]Test chamber height = 14 feet.

Design Criteria For NCEL Cold Air Jet Destratifier

NCEL-designed cold air jet destratifier is based upon the ability of an air jet to entrain surrounding air and throw it across large distances. These principles are well known and are documented by ASHRAE (Ref 7 and 8). Figure 33 is a nondimensionalized drawing of the destratifier which, when used with the NCEL cold air jet destratifier design parameters presented in Appendix B, can be used to design a destratifier for most hangars. The equations used for the destratifier system design are based upon principles stated in References 7 and 8 and are as follows:

$$Q_d = 0.00278\ V/N_d \qquad (30)$$

where: Q_d = destratifier flow, ft^3/min

N_d = number of destratifiers to be installed

V = volume (hangar), ft^3

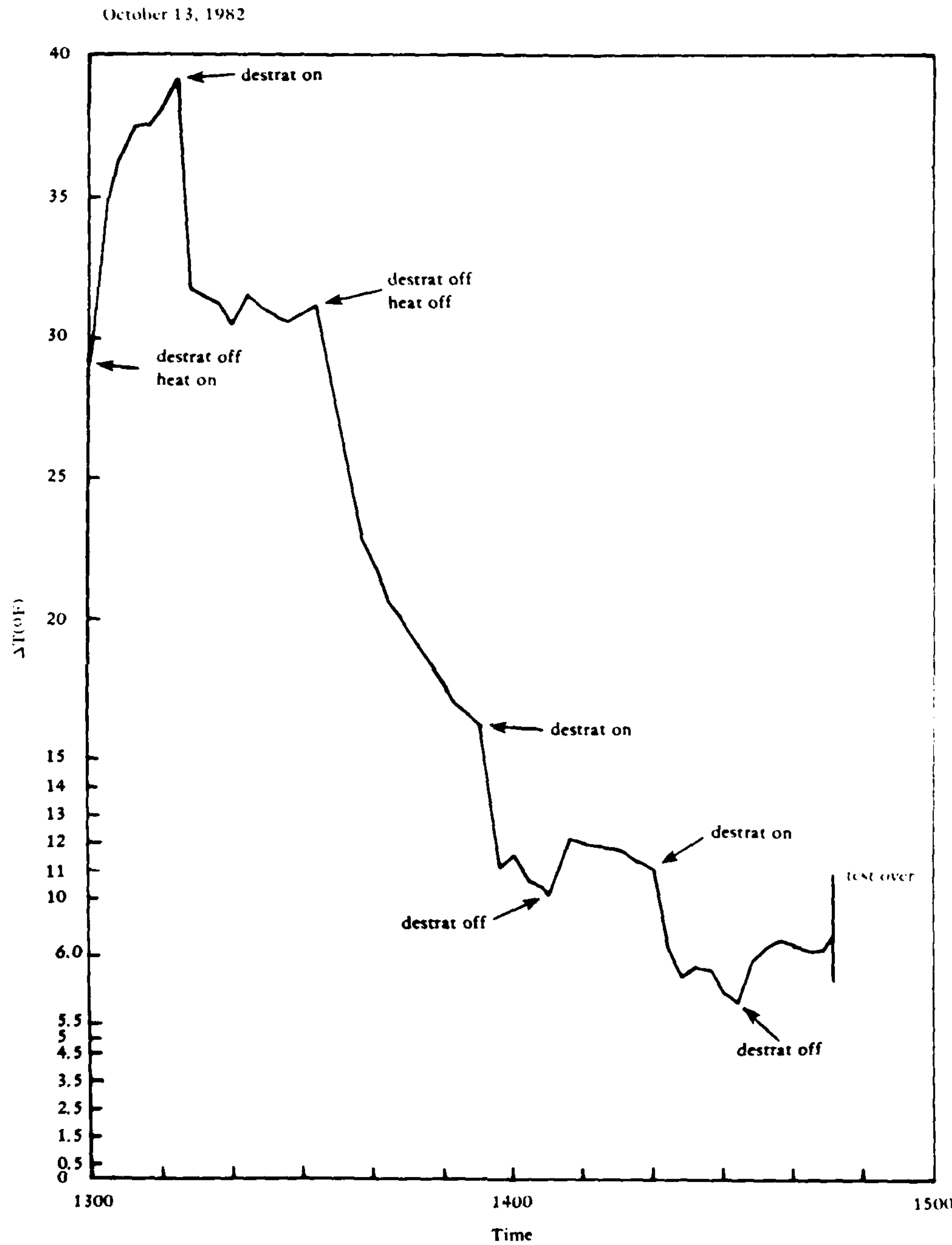


Figure 31. Destratification tube test chamber results.

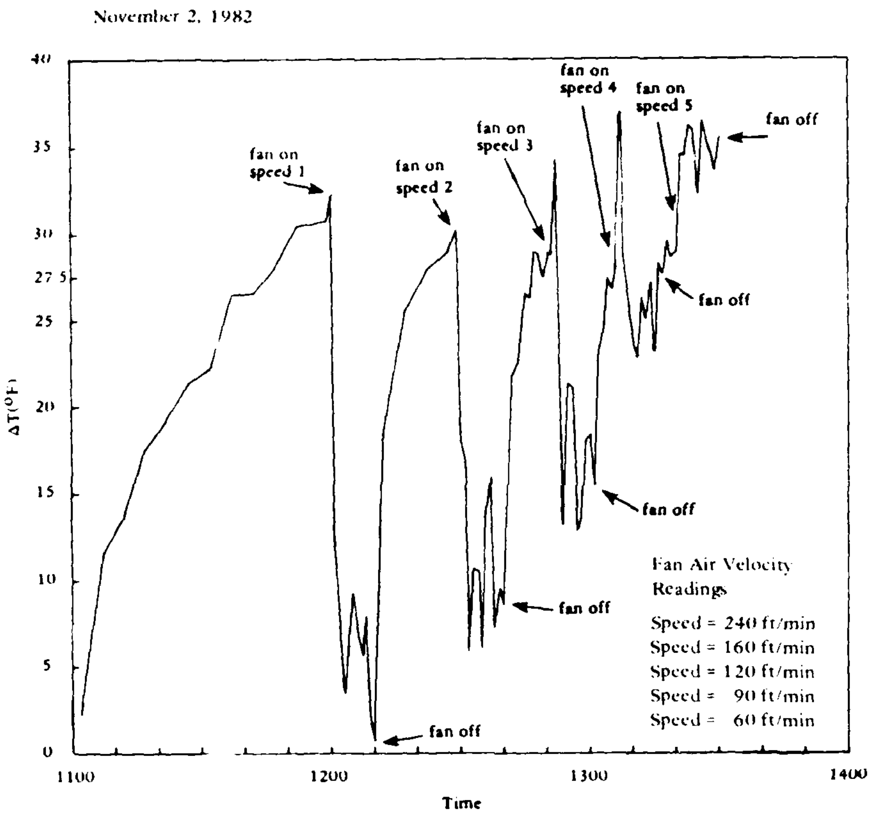


Figure 32. Ceiling fan test chamber results.

$$U_V = 0.022U_r^2 \, W_f^2/Q \tag{31}$$

where: U_V = air exit velocity at destratifier nozzle, fpm

U_r = residual air velocity at distance W_f, fpm

W_f = width of hangar, ft

$$dia_i = 24 \, (Q_d/\pi U_v)^{1/2} \tag{32}$$

where dia_i is the nozzle diameter in inches.

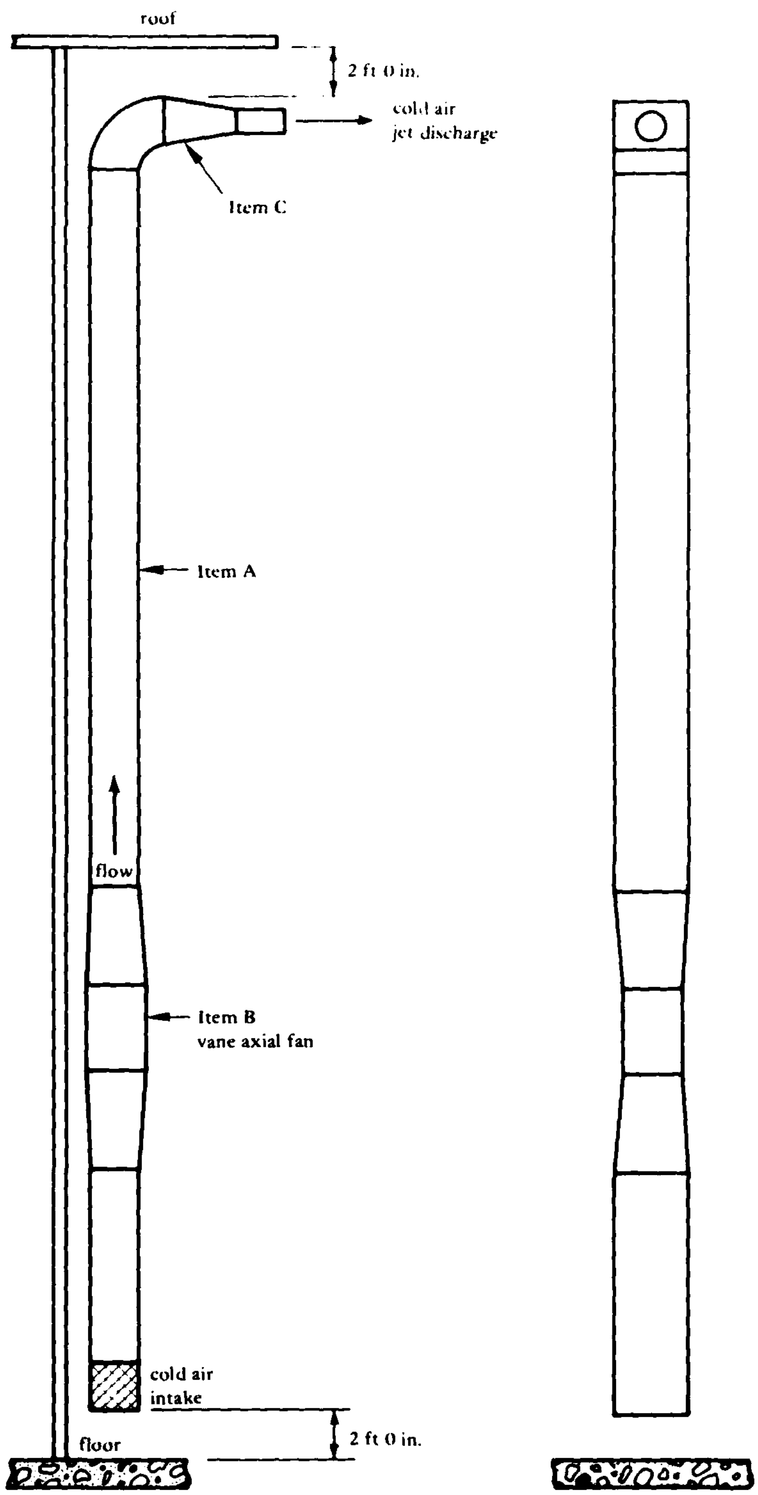


Figure 33. Cold floor air jet injection destratifier.

For this example, a residual air velocity of 170 fpm was used with a minimum air entrainment ratio of 20; and the hangar's width was used as the throw distance for the hangar. The design data for nozzle diameter, presented as Appendix A, should be rounded to the nearest 1/4 inch. The recommended number of units is based upon providing one air change per hour. Hangars, however, have their overhead areas sectionalized by draft curtains to curtail the spread of smoke and flame in case of fire. In some instances, the number of overhead sections will exceed the number of destratifier units recommended. In such circumstances, additional destratifiers are required so that at least one destratifier is located within each section. Some older hangars have draft curtain designs which sectionalize the overhead like an egg crate. Such design will prevent the cold air jet destratifier from being effective.

Installation And Operational Evaluation Of Destratifiers

Destratifiers based upon the NCEL, LANTDIV, and cold air blower concepts were installed in Navy hangars for evaluation. The NCEL-designed destratifier was installed at the Navy Air Rework Facility (NARF), Norfolk, Va., in one of two bays in hangar V-147. Steam consumption in both bays was measured; thermostat settings for both bays was kept at 65°F. Table 13 presents the reduction in steam consumption the destratified bay versus the stratified bay. It was determined that the destratified bay consumed 29% less heating-related energy, based upon measurements made during parts of two heating seasons.

The LANTDIV heating system modification was installed in the center section of a hangar located at the Naval Air Development Center (NADC), Warminster, Pa. (Figure 34). Draft curtains across the hangar divided the overhead area into three sections of equal volumes and provided a solid barrier 15 feet deep from the roof down toward the floor. Thermocouple arrays were placed in two of the three sections (one with the LANTDIV modified destratifier and one without). Thermocouples for each section were placed at: (1) ceiling level directly above the hot air blower; (2) hangar centerline at ceiling level--20 feet away from the hot air blower; (3) hangar centerline--2, 4, 8, and 12 feet below the ceiling; (4) center wall--1 and 10 feet above the floor.

Hourly data were obtained for 3 months and recorded on a data logger. Table 14 presents a synopsis of the data. The LANTDIV heating modification resulted in an average decrease in the destratified section of 2°F temperature.

Three commercial cold air floor blowers were installed according to the manufacturer's recommendations in a hangar also located at NADC Warminster (Figure 35). A thermocouple array was installed at the following locations: (1) on the hangar centerline--at the ceiling level and 2, 4, 8, and 12 feet below the ceiling; (2) 1 and 10 feet above the floor; and (3) outside.

Table 13. Comparison of Destratified (NCEL Concept) and Stratified Hangar, Building V147, NARF Norfolk

[Electric power consumption: 240VAC @ 3.2 amps/unit (7 units x 240 x 3 x 2 x 24 hr/day x 35 day/1,000 = 4,516 kWh); central steam plant efficiency = 68%; electric generation heat rate = 11,600 Btu/kWh; steam savings = 30%; net energy savings = 29%.]

Date	Temperature[a] (°F)					Steam Consumption[a] (MBtu)		Steam Savings[a] (MBtu)
	Outside	Destratified Bay		Stratified Bay		Destratified Bay	Stratified Bay	
		Floor	Ceiling	Floor[b]	Ceiling			
2/18/82	54	66	74	65	80			
2/19/82	54	66	80	65	83	213	292	79
2/25/82	47	65	74	65	86	889	792	<97>
2/26/82	35	69	74	65	86	280	430	150
3/3/82	42	69	74	65	82	999	1,259	260
3/5/82	44	68	74	65	82	298	229	<69>
3/8/82	32	67	76	65	88	341	715	374
1/14/83	48	68						
1/18/83	51	69				138	621	483
1/20/83	43	65				244	606	362
1/24/83	48	71				1,447	2,076	629
1/27/83	46	52				545	443	<102>
1/31/83	50	66				672	1,213	541
Total						6,066	8,676	2,610

[a]Blanks indicate data not available.
[b]Temperature not measured, assumed to equal thermostat setting.

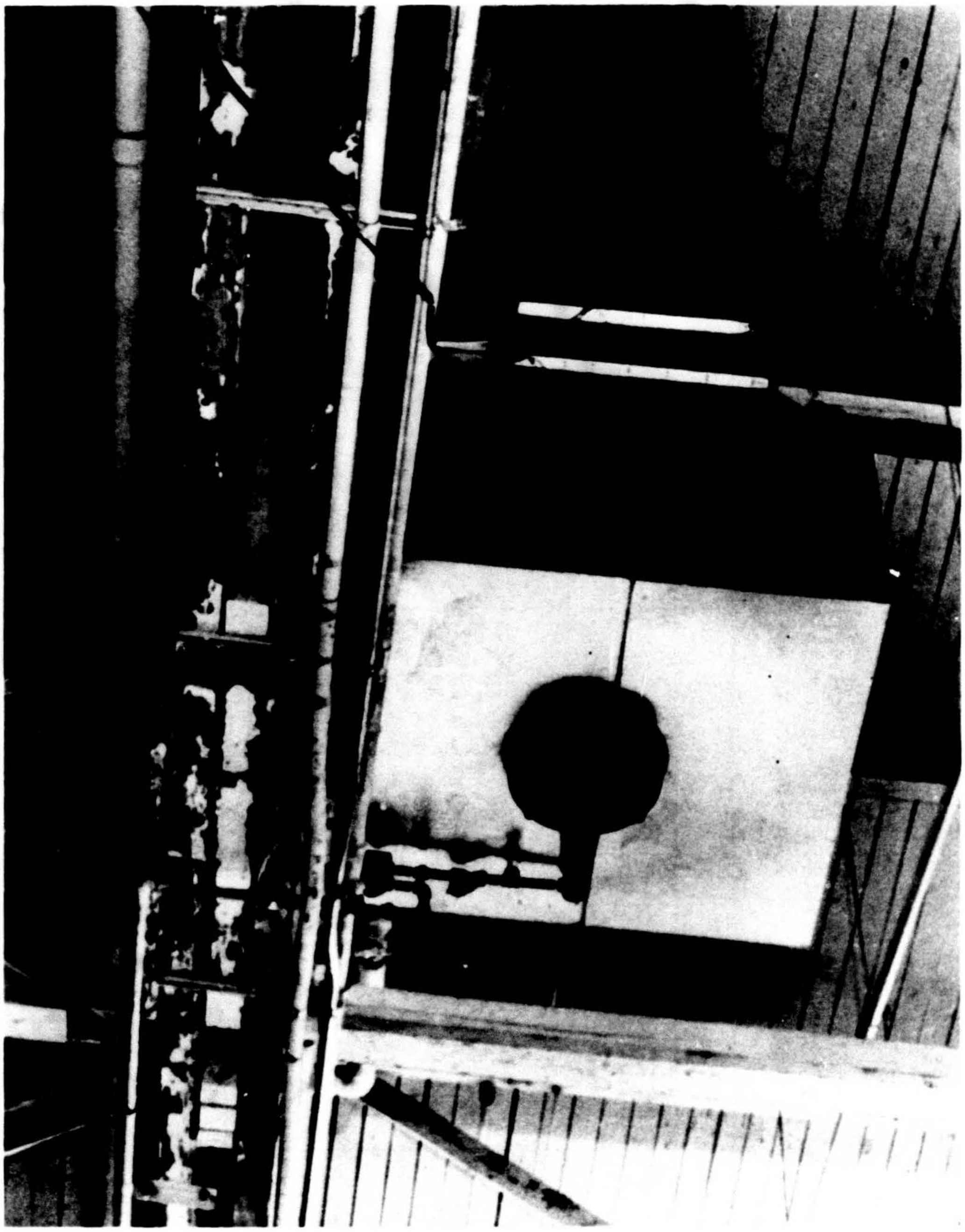


Figure 34. LANTDIV modification installed, NADC Warminster.

Table 14. Temperatures Taken Before and After LANTDIV Modification at NADC Warminster

Temperature (°F)			
Ceiling Bay 2	Ceiling Bay 3	Floor	Outside
Before Modification[a]			
61	65	51	34
61	65	53	33
73	69	58	34
78	72	58	33
78	75	60	33
79	74	59	34
77	75	59	33
78	75	59	33
79	76	60	34
79	76	60	34
76	75	58	33
74	73	58	34
70	70	55	29
72	71	56	30
75	72	57	29
74	72	58	30
76	73	60	29
81	77	64	30
74.5[b]	72.5[b]	57.6[b]	32.7[b]
After Modification[c]			
68	70	55	43
69	70	56	30
72	72	58	30
72	72	58	30
73	73	57	29
75	75	58	29
75	75	58	29
77	75	58	30
75	75	58	30
76	75	59	34
78	77	60	36
79	78	63	38
80	79	65	40
81	79	64	43
81	80	65	46
74	75	61	46
61	64	53	36
64	66	55	36
73.9[b]	73.9[b]	58.8[b]	33.7[b]

[a] Ceiling 2/floor ΔT_c = 16.9°F; Ceiling 3/floor ΔT_c = 14.9°F; Average ceiling ΔT_c Bay 2/3 = 2°F.

[b] Average.

[c] Ceiling 2/floor ΔT_c = 15.1°F; Ceiling 3/floor ΔT_c = 15.1°F; Average ceiling ΔT_c Bay 2/3 = 0°F.

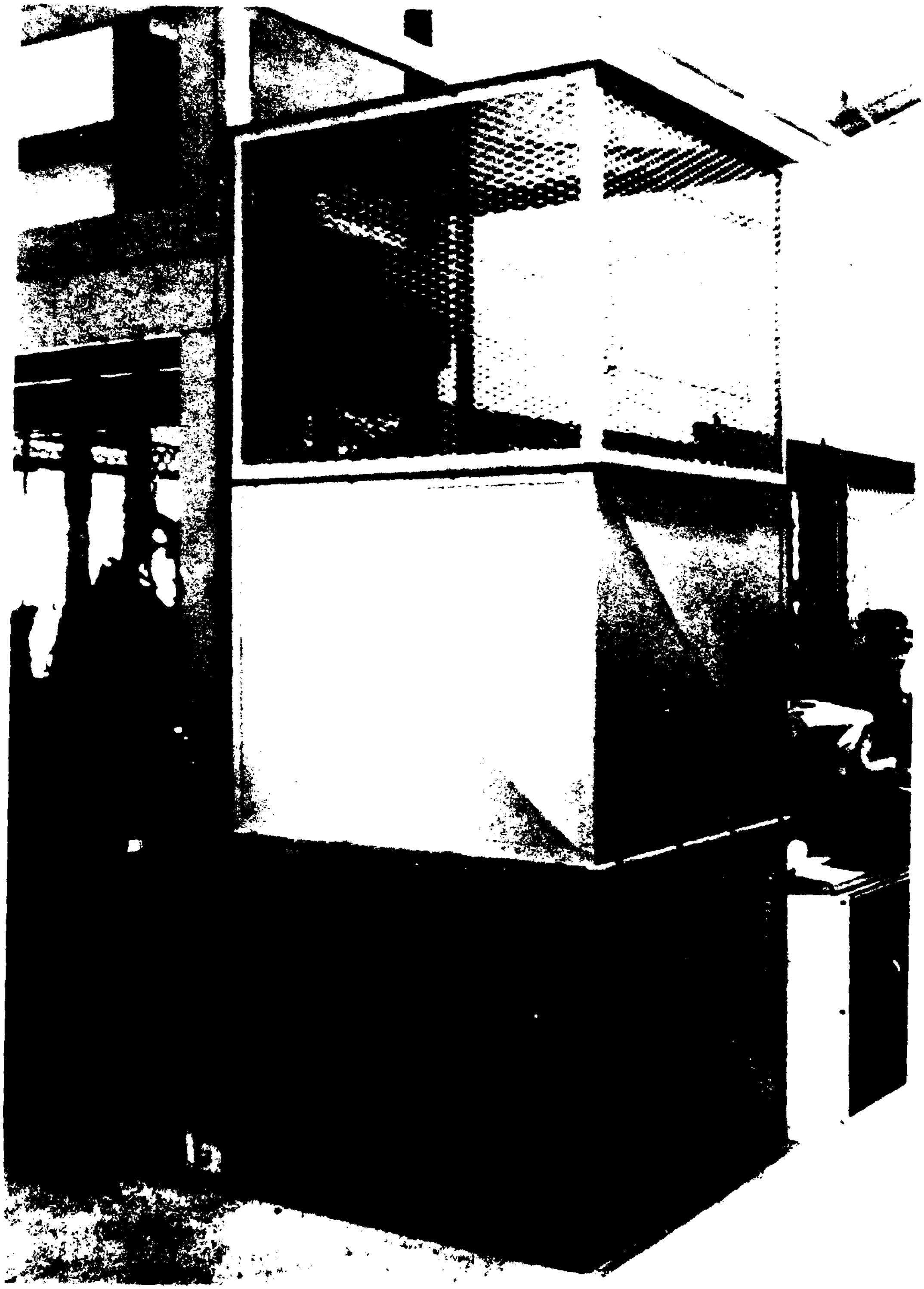


Figure 35. Cold air blower installed, NADC Warminster.

Temperatures were recorded hourly on a data logger during the 1982/83 winter. Data were obtained with and without the cold air blower in operation. A synopsis of the data is presented in Table 15 and Figure 36. The data did not show that the commercial cold air blower was effective. One reason that this commercial unit was effective at tests conducted at NCEL and not in the hangar could be related to the hangar's configuration (height and width) coupled with the inability of the units to move air over distances typically existing in hangars. However, no data was obtained to substantiate this conclusion.

Results

Of the five destratification concepts evaluated, only the NCEL cold jet destratifier produced meaningful results and is recommended for new and existing hangars.

The cold air floor blower, the only commercial concept actually evaluated in a hangar, performed very well in a 25-foot high building with the unit placed in the building's center. Hangar installation would require that the units be placed along the structure's wall. Because of hangar width, height, and volume and operational usage characteristics, the cold air blower is either not adaptable to hangar applications.

The LANTDIV heating system modification had a destratification efficiency of 11% and did save energy. Its cost ($3,500/heater) is not much less than the more efficient (29%) NCEL unit ($8,000/unit) when the number of heaters are considered, and therefore it is not recommended for retrofit.

RADIANT HEATING

The principles of radiant heating, guidelines for optimum system design, and information on physiological effects are presented in this section of the report. Cost analysis procedures are provided for evaluating and comparing the cost-effectiveness of radiant heating with other types of heating systems. Also presented are the measures required to comply with the National Fire Protection Association (NFPA) codes.

Principles Of Radiant Heating

High-intensity, porous, refractory, infrared (IR) radiant heaters (Figure 37) are best applied where and when convective heaters are impractical. IR radiant heaters emit intense heat in a straight line-of-sight direction, which makes them impractical for applications in areas below 8 feet in height because, in such areas, hot spots and uneven heat distribution result. Areas shaded from the IR radiant heat do not receive the same heat flux density as areas exposed to the IR radiation.

Table 15. Temperatures Taken With Commercial Cold Air Floor Blower, NADC Warminster

Temperature (°F)				Floor/Ceiling Temperature Difference (°F)
Ceiling	Floor	Outside	Floor/Outside	
Destratifier Off				
76	61	34	27	15
76	61	34	27	15
76	61	34	27	15
76	61	34	27	15
76	61	33	28	15
78	68	35	33	10
79	66	27	39	13
82	75	29	46	7
84	73	33	40	11
81	71	29	42	10
81	73	27	46	8
76	67	20	47	9
73	62	18	44	11
83	74	26	48	9
89	79	33	46	10
83	75	33	42	8
83	72	44	28	11
82	72	41	31	10
77	69	45	24	8
71	62	48	14	9
73	64	44	20	9
74	68	43	25	6
75	68	43	25	7
74	67	50	17	7
73	66	57	9	7
71	63	46	17	8
Destratifier On				
75	69	52	17	6
76	64	44	20	12
79	66	38	28	13
72	68	51	17	4
74	66	43	23	8
73	62	44	18	9
75	64	44	20	11
76	64	44	20	12
77	65	43	22	12
77	65	36	29	12
78	65	36	29	13
79	65	36	29	13
80	68	43	25	12
81	69	43	26	12
76	64	43	21	12
71	67	48	19	4
74	69	50	19	5

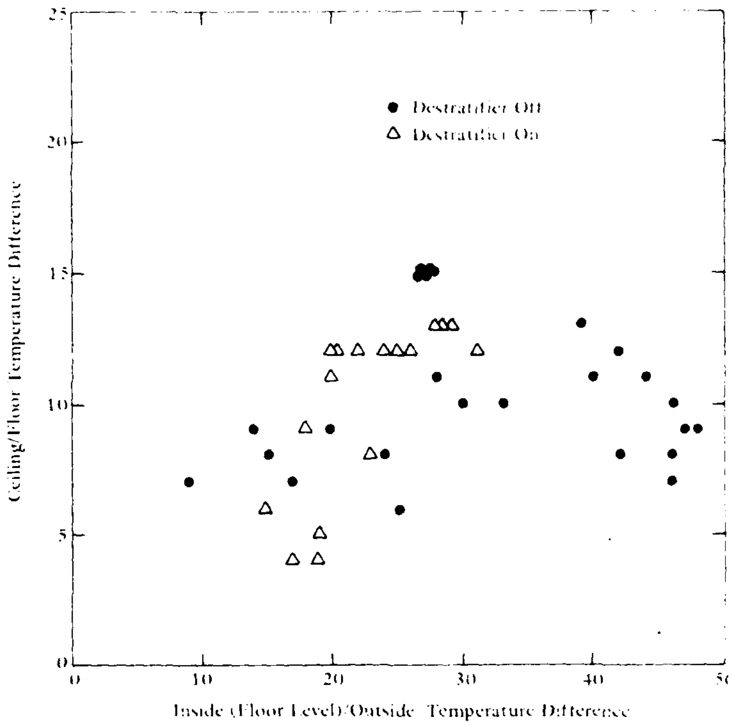


Figure 36. Cold air blower performance, NADC Warminster.

Figure 37. High-intensity porous refractory IR radiant heaters.

IR radiation, like light energy, is transmitted in the form of invisible electromagnetic radiation and does not require a medium to transfer heat.* IR radiation is the primary mode of heat transmission; all matter, whether solid, liquid, or gas, above the temperature of absolute zero, emits IR radiation. Upon striking an absorbing mass, IR radiant energy is converted into heat, and this absorptive mass (e.g., a concrete floor) becomes a reservoir for heat storage. Once the floor temperature is greater than the interior ambient air temperature, this heat reservoir gives off heat to the surrounding environment by convection, conduction, and radiation. Quick heat recoveries are possible because the hangar is heated from the floor up rather than from the ceiling down, as is normal with convective heaters.

The performance of an IR radiant heater depends upon the emissivity of the heater's emitter surface and the absorptivity of the receiver surface. The emissivity of the emitter determines the output efficiency of the emitter while the absorptivity of the receiver determines at what efficiency percentage the heat is being absorbed. For any given material associated with hangars, the emissivity and absorptivity factors can be considered to be the same. Either surface factor is determined on a scale from 0 to 1.00 where 0 means no emission of any IR radiation and 1.00 means emission of all its radiation. Realistically, neither can be achieved. Reflectivity supplements absorptivity, as shown in Figure 38.

The emitter's surface temperature determines the wavelength and intensity of the radiant energy: the higher the surface temperature, the shorter the wavelength and the greater the intensity. The intensity, q (or rate of heat transfer), is directly proportional to the fourth power of the surface temperatures, as shown in Equation 33:

$$q = K \varepsilon a \theta^4 \tag{33}$$

where: q = rate of IR emission or emissive power, Btu/hr

k = $0.17^3 \times 10^{-8}$ Btu/hr ft^2°R^4 (Stephan-Boltzman constant)

ε = emissivity of the emitter surface, dimensionless

a = surface area of the emitter, ft^2

θ = absolute temperature of the emitter surface, °R

*Air, the medium for convective heat, is a poor absorber of IR radiation.

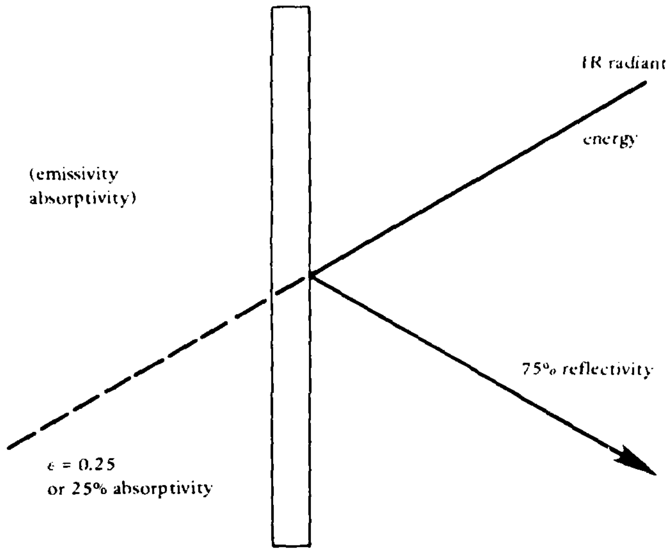


Figure 38. Reflectivity, the supplement of absorptivity.

Advantages Of Radiant Heaters

A properly designed and installed IR radiant heating system in an aircraft hangar can eliminate many of the heating problems that have plagued hangar managers. Gas-fired, high-intensity, porous, refractory radiant heaters are recommended for hangar heating because of:

- Lower energy cost - Since radiant heaters do not heat the air, no fans are needed to distribute the heat, thus saving capital cost of fan components. With no moving parts, chance of equipment breakdown is reduced.

- Lower ambient air temperatures for the same thermal comfort provided by convective heaters - Since radiant heaters do not use air as the medium to transfer heat, hangar air does not need to be heated directly. Thus, heat normally used to heat air is diverted to heating the floor and objects, which heat up faster than the ambient air and then emit heat to the air. Heat, therefore, is emitted by conduction, convection, and radiation with radiation applied directly to personnel, the primary targets.

- Less building heat loss - Since less heat is contained in the hangar air volume, less heat is being transmitted to the walls and roof.

- Less air stratification and air infiltration - Lower ambient air temperatures allow easier control of the air volume and reduce stratification. The reduced temperature difference between the inside and outside air reduces the air infiltration losses through gravity ventilation or stack effect.

- Zone heating - Zones can be heated independently for changing work requirements. Rather than heating the whole hangar, work areas may be fully heated while storage areas may be heated to just above the dew point or freezing temperatures. Unoccupied areas may be left unheated; however freeze protection is usually required. Zone heating requires additional heaters and thermostat controls.

- Heat storage - The radiant energy striking the concrete floor is converted into heat, which is absorbed by the floor; the floor thus becomes a heat storage reservoir. In addition, since the floor surface heats more quickly than the ambient air temperature, the floor acts as a radiator and gives off heat. Even when the hangar doors are opened to the cold outside air, the radiant heaters continue to store heat in the floor. When the doors are closed again, this stored heat plus the direct heat from the radiant heaters enable the temperature at the floor level to recover quickly (15-20 minutes) for thermal comfort.

- Condensation/corrosion control - If humidity or condensation is a problem, radiant heaters can supply just enough heat to keep the floor or an object just above the dew point or freezing temperature. This helps eliminate unwanted moisture and corrosion problems.

Designing An IR Radiant Heating System

Several factors must be carefully considered when an IR radiant heating system is being designed for any building. As indicated earlier, a properly designed and installed heating system in an aircraft hangar can eliminate many operational and cost problems for the hangar manager.

Emissivities. The critical factor in designing an efficient IR radiant heating system is the matching of the emissivity of the heater to the absorptivity of the objects to be heated (i.e., personnel, concrete floors, or other items). The emissivity of a material may vary under different temperatures and IR wavelengths. For example, the emissivity of white paint (ZnO) is approximately 0.18 (18% absorptance) for a solar light IR wavelength of 0.6 micron and 0.95 (95% absorptance) for IR radiation of 9.3 microns at 100°F. Appendix A in Reference 9 has a list of emissivities for building materials exposed to various heat temperatures and wavelengths. To maximize the thermal efficiency of a radiant heating system, the IR radiant heaters need to emit IR energy in the range that the objects to be heated will most readily absorb.

National Fire Protection Association (NFPA) Codes. The purpose of the NFPA regulations is to insure that the heaters are safely installed and will not create a hazard or interfere with the existing fire protection systems.

Volume 10 of Reference 10 indicates that heaters employing an open flame or glowing element that are listed for use in aircraft hangars may be installed if they meet the spacing requirements in that chapter. Further information on installation requirements for radiant heaters can be found in the NFPA Codes in References 10 through 15.

Clearances

IR radiant heaters will most likely be suspended from the ceiling or roof structural members to avoid contact with moving cranes or aircraft with high tail sections. These heaters cannot be installed above fire detection or sprinkler systems as the intense heat may activate these systems. Also, if the units are to be suspended among the roof structural members, the path of the IR radiant heat must be unobstructed to the floor level. Structural members, wiring, pipes, and other items can be damaged by the intense heat. Figure 39 shows the damage that can occur when an IR radiant heater is installed just above a structural member; the intense heat charred the paint and exposed the metal to heat stress and corrosion.

Figure 39. Damage to structural member caused by IR radiant heater.

Layout

Two layouts--perimeter and checkerboard--are possible.

1. Perimeter. A perimeter layout of radiant heaters such as that in Figure 40 will heat the hangar from the base of the wall to the center floor. IR heaters should be oriented so that their field of coverage starts at the floor edge with the wall and continues toward the center (Figure 41). To minimize heat loss through the walls, the heater's line-of-sight should not include the walls. If the hangar is extra large, an additional center row mounting may be necessary. Additional heaters will be required over the main hangar doors as shown in Figure 42 because of the larger heat loss when the doors are open.

2. Checkerboard. An overhead checkerboard layout would evenly distribute the heaters throughout the hangar (Figure 43). However, this layout would require more heaters than the perimeter layout and would increase the network of pipes and electrical wiring.

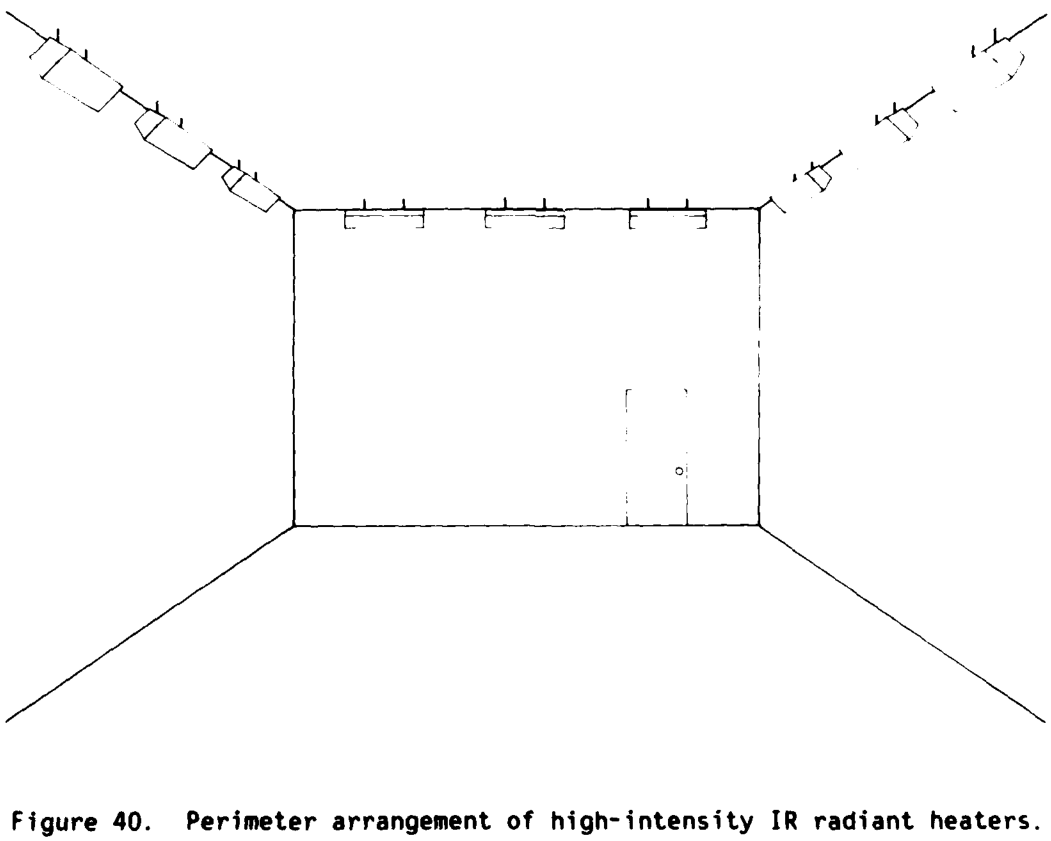


Figure 40. Perimeter arrangement of high-intensity IR radiant heaters.

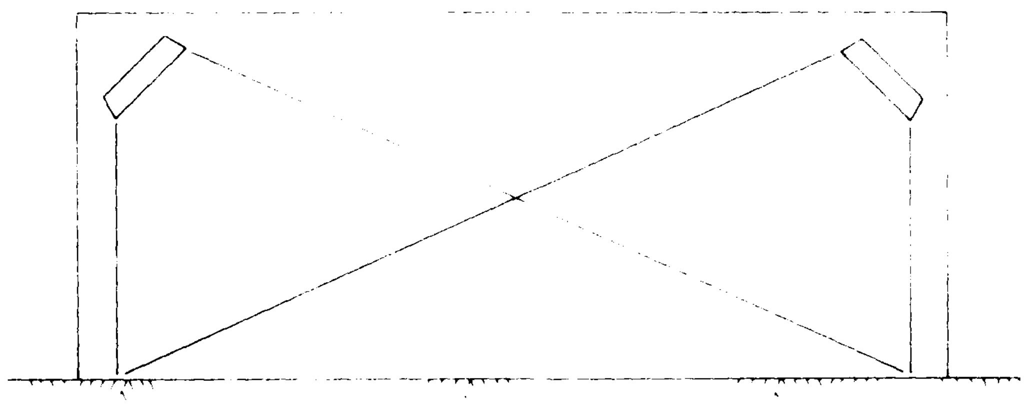


Figure 41. Recommended IR radiant heaters line-of-sight coverage.

Figure 42. Additional heaters required over hangar doors.

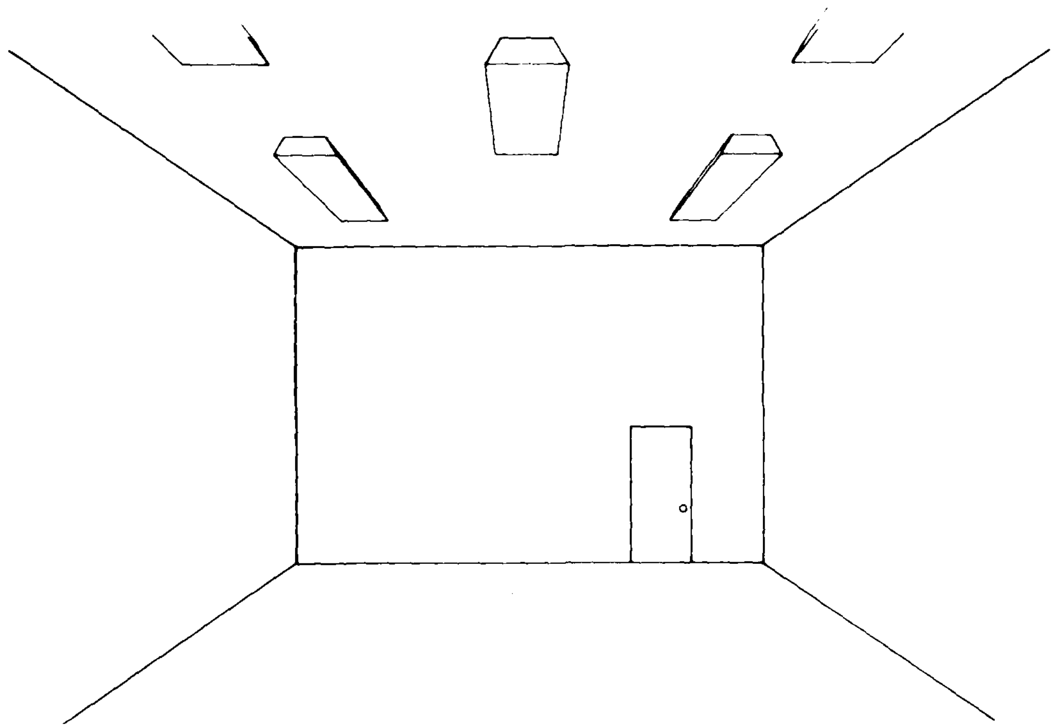


Figure 43. Checkerboard arrangement of high-intensity IR radiant heaters.

Venting

Venting is required for fossil-fuel-fired radiant heaters. Building materials contiguous to the exterior (e.g., glass skylights) are potential collection points for condensation from warm moist air or from flue gases. If the outside surface temperature is below the dew point temperature, condensation can form and drip down on the floor below, which is unacceptable for aircraft maintenance and personnel safety.

Also, products of combustion from fuels such as LPG and propane mixed in with natural gas may have a higher sulfur content. When these are combined with water vapor, these products can form sulphuric acid. Venting of fossil-fuel-fired heaters will remove the combustion gases and prevent condensation.

Two types of venting are discussed below.

1. Direct. Direct venting of the heaters will remove flue gases from the hangar through exhaust stacks in the roof (Figure 44). If such venting is used, these stacks should have dampers at the top to prevent downdrafts. These dampers prevent the IR burners from being "blown-out" by downdrafts and help eliminate the funneling of cold air down through the stack when the heaters are not in use. The exhaust stacks must rise directly from the IR radiant burners (Figure 45). Because a horizontal stack will impede the rise of hot flue gases, an incline in a horizontal stack will allow the gases to rise. If two vent stacks are joined, the diameter of the resulting stack must be larger than the feeding stacks to handle the combined flow rate from the feeding stacks.

Figure 44. Venting of burners through the roof via exhaust stacks.

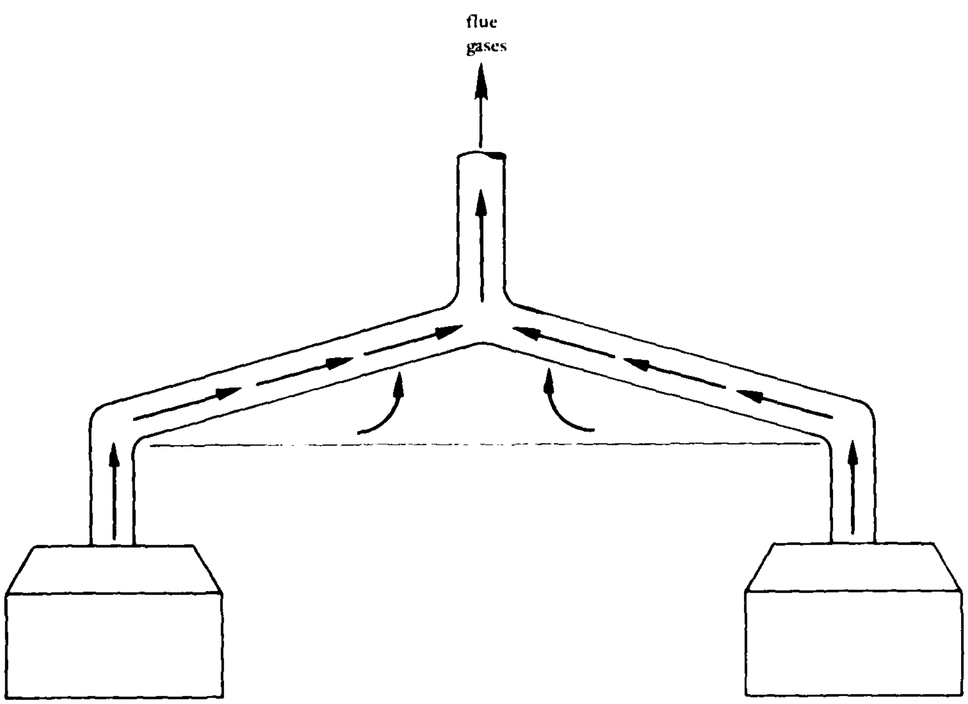


Figure 45. Incline of exhaust stacks to be sure of continuous rise from the IR radiant burners.

2. <u>Exhaust</u>. Another venting method is the use of exhaust fans with spring-loaded draft shutters installed in the ceiling (Figure 46). When the exhaust fans are operating, positive air pressure forces the draft shutters to open and exhaust the flue gases. When the exhaust fans are shut off the force of the loaded spring on the draft shutters forces them closed preventing warm air from escaping and cold air downdrafts from entering. Exhaust fans can be controlled automatically either of two ways:

- A relay switch can turn on the exhaust fan whenever the heater operates and can turn it off whenever the heater is turned off. A timed delay of several seconds or minutes can be included to delay the fan shutoff to ensure purging of the flue gases.
- The on-off function of the fans can be automatically controlled by a humidity control device. That is, whenever the moisture content from the flue gases reaches a sensor's preset level, the humidity control automatically turns on the exhaust fans. Once the moisture content of the air drops below the lower preset moisture range, the exhaust fans shut off. These sensors could be placed at several points in the roof structural members.

A manual override switch included in the above circuits could be used to bypass the above two systems should the need arise (e.g., to vent smoke or odors).

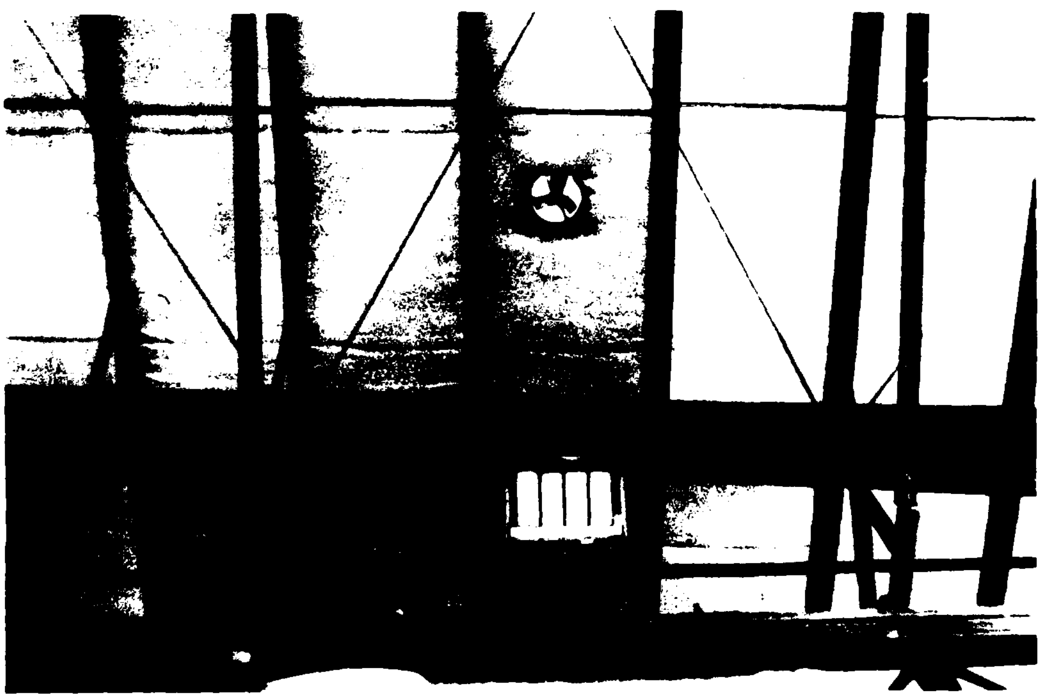
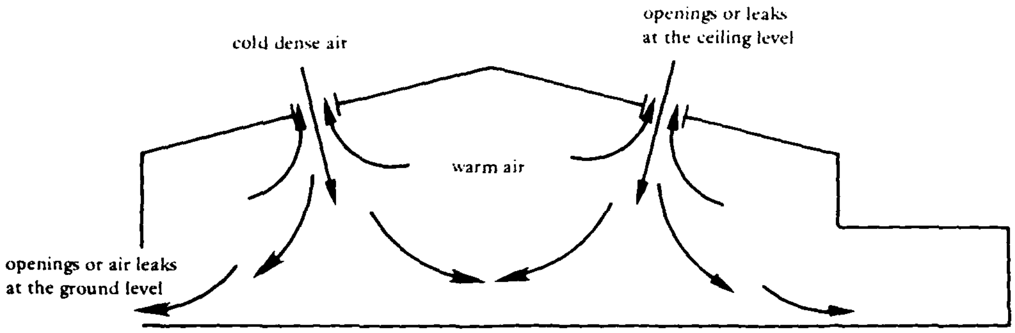


Figure 46. Exhaust fans fwith spring-loaded draft shutters used to vent the flue.

Minimizing Building Air Leakage

When the air temperature inside the hangar exceeds the outside air temperature, potential difference in the density levels between the warm and cold air can cause gravity air infiltration (Figure 47). Warm light air rises and forces itself out of the building through cracks and static openings; cold dense air is forced down by gravity to replace this warm air. The larger the temperature difference, the greater is the air leakage potential. To minimize the effects of gravity air infiltration, heater flue gas stacks or vents must have dampers which close when the unit is not operating.



Requirements:

1. Outside air temperature $<$ inside air temperature.
2. Ground level openings $>$ ceiling level openings.

Figure 47. Gravity ventilation.

Heater Ignition

Electronic glow coils or electronic spark ignition are recommended to ignite gas burners. They are very reliable and provide additional safety to prevent an accumulation of gas fumes (such as when a pilot light gets blown out). These ignition systems have built-in safety features to shut down the fuel supply if the spark fails to ignite the burner in a specified number of seconds. Elimination of pilot lights saves additional energy as well as the inconvenience of re-lighting pilot lights if they blow out.

Thermostats

Thermostatic controls must be shielded to prevent heat buildup in the controls, which are installed below the IR radiant heaters. Thermostats must be located outside the line-of-sight of the radiant energy.

Maintenance Access

Section 9-1.2.3 of Reference 10 states that access to suspended heaters shall be provided for recurrent maintenance. Though a truck with a hydraulic lift may fulfill this requirement, catwalks with an access ladder is a better alternative. Catwalks are more expensive initially, but will give immediate and assured access to heater units at any time and for any hangar floor configuration. With catwalks, the direction of the heater may be changed easily and at any time to meet changing heating requirements at the floor level.

Electric Solenoid Shutoff Valves

Installation of an electrical solenoid shutoff valve in the fuel supply line to each heater (Figure 48) is recommended to isolate each heater. With this installation, other units on the same fuel line can be used while an isolated unit is awaiting maintenance. In another example, if a tall tail section of an aircraft is placed too close to a heater, the switch can be shut off to the affected heater to prevent heat damage to the aircraft. The shutoff switch can be located directly below the heater, at a central control point, or at both locations.

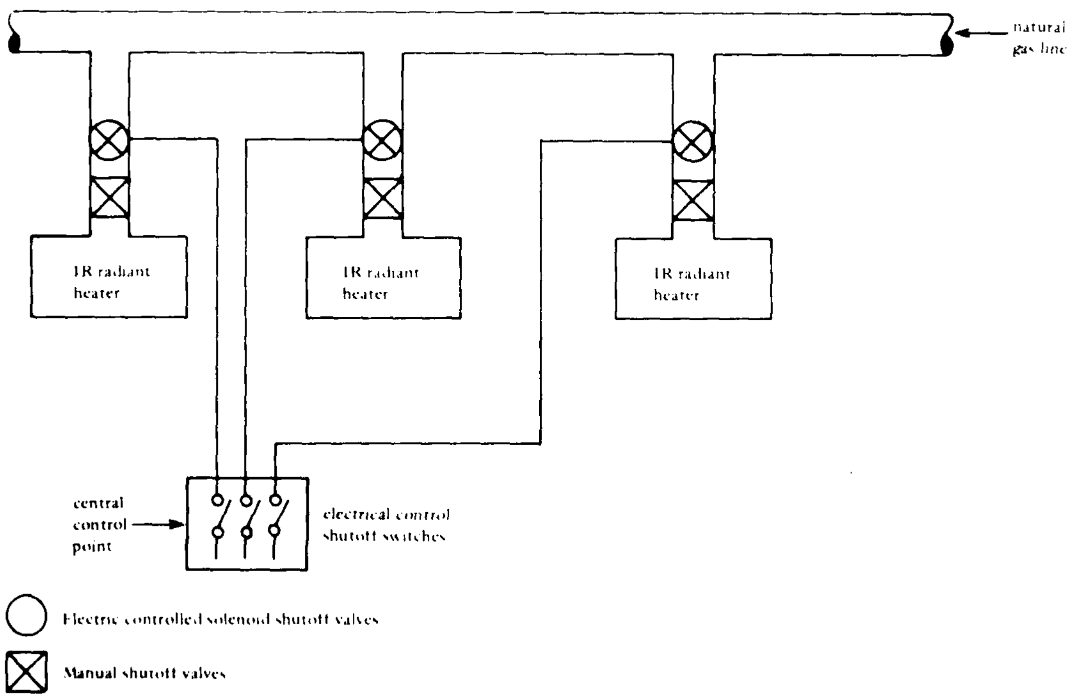


Figure 48. Individual and central shutoff valves to isolate each heater.

Manual shutoff valves for each heater are required (NFPA regulations, Ref 10 through 15). Even with catwalks in place, a knowledgeable person would need time to reach the heater, and time may no' always be available. Therefore, another consideration is that loss of one heater may not degrade a heating system, but the loss of several may degrade it if several units are on the same fuel line and do not have separate shutoff solenoids.

Certification By Manufacturer

A survey was conducted to determine the experience of engineering organizations (government and civilian) in designing heating systems. The survey indicated knowledge of radiant heating system design is not widespread.

It is recommended that the contract for installation of a radiant heating system specify that the construction drawings for the system design be reviewed by the product's manufacturer to insure compatibility of his heater with the proposed installation. The manufacturer should follow up with an on-site certification of the installed system. Though this may increase the installation cost slightly, it will help eliminate costly correction of design errors. Some examples of design errors in existing radiant heating systems are:

- High intensity IR heaters installed just above roof structural members (Figure 39).
- Vent stacks installed horizontally (Figure 44), which can cause the flue gases to stagnate.
- Vent stacks installed without downdraft dampers, resulting in cold downdrafts.

Physiological Effects Of IR Rad.ation On Personnel

Physiological concerns as a result of use of IR radiant energy are those adverse effects that may damage eyes and skin of personnel.

Eyes. The eye normally has two forms of protection from excessive exposure to IR radiation energy: (1) the iris naturally contracts and (2) photophobia or painful reaction to intense light occurs. However, the lens of the eye can focus concentrated IR radiant energy onto the retina. Damage to the eye in the form of cataracts and retinal burns can occur if the iris does not have enough time to contract and the intensity of the IR radiant energy is great enough (Ref 16). The eye does not have a circulatory system to act as a cooling system to dissipate heat quickly (Ref 17).

In various studies it has been shown that short wavelengths of IR energy in the range of 1.0 to 2.0 microns can cause cataracts of the eye after prolonged and intimate exposure, but this exposure must be intense and for a prolonged period of time for the damage to occur. Eye damage did not occur at IR radiant energy wavelengths longer than 2.0 microns, which is the range of typical radiant heaters.

<u>Skin</u>. Studies documented in Reference 18 show that skin absorption is sensitive to the wavelength of the IR radiation; the skin absorptance is variable in the low range between 0.4 to 2.0 microns (Figure 49). For wavelengths from 2.5 to 20.0 microns the skin absorptance of IR radiant energy is approximately 97 to 99% (Ref 18). For IR wavelengths longer than 2.6 microns, the skin, regardless of color, is essentially a black body. IR energy penetrates to between 0.0008 and 0.00012 inch beneath the skin surface and, thus, interacts directly with the nerve endings and small blood vessels. This interaction gives the sensation of warmth.

The skin is more sensitive to the longer IR wavelengths than the shorter IR wavelengths. Also, objects with high moisture content absorb longer (>2 microns) wavelength energy more readily and at a higher percentage. Thus, smaller amounts of longer IR wavelength energy are needed to produce a sensation of warmth.

In summary, no adverse health effects associated with IR radiant energy have been detected with wavelengths longer than 2.0 microns. The skin is more responsive to these longer wavelengths for heat absorptance and warmth sensation.

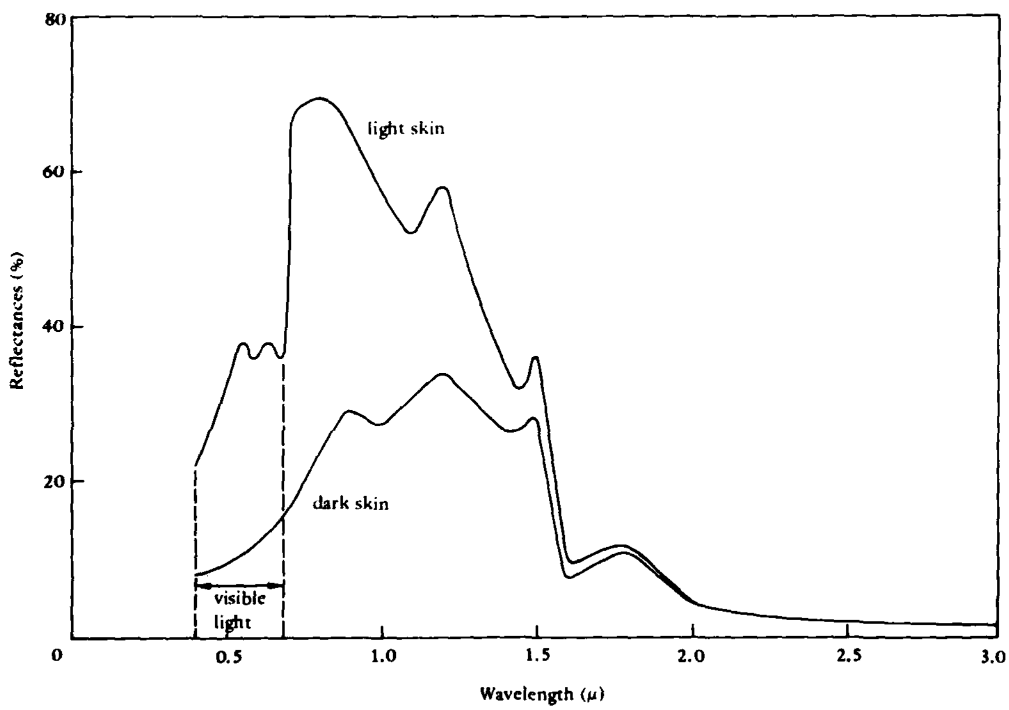


Figure 49. Average special reflectance values for human skin.

Comparative Cost Analysis

Before installing an IR radiant heating system, the designer should prepare a cost analysis on the existing heating system and on any proposed IR radiant heating system. An estimate can then be made on the seasonal fuel cost for each system. This fuel cost can be estimated by using the degree-day method presented by the American Society of Heating, Refrigeration, Air Conditioning Engineers (ASHRAE). This estimate would be based upon average weather data collected for the locality (NAVFAC P-89, Engineering Weather Data for Degree Days).

$$\text{Annual Fuel Consumption Units} = \frac{24\,(H_L)(\text{degree days})}{\eta\,(T_i - T_d)\,\eta} \qquad (34)$$

where: H_L = calculated total building heat loss, Btu/hr

η = heating system efficiency

T_i = inside design temperature, °F

T_d = outside design temperature, °F

Once these estimates are made, then a cost comparison can be made to determine if the payback period will justify the cost of retrofitting a building with an IR radiant heating system. Projected annual savings can be used to determine the payback period of the retrofit using the following equations:

$$\text{AHC} = \text{AFCU} \times \text{fuel cost/unit} \qquad (35)$$

$$\text{PCS} = \text{AHC}_{\text{convective}} - \text{AHC}_{\text{IR}} \qquad (36)$$

$$\text{PBP} = \frac{\text{Installation cost of radiant heater}}{\text{PCS}} \qquad (37)$$

where: AHC = annual heating costs, $

AFCU = fuel units (i.e., gallons, Btu's)

PCS = project cost savings for fuel per year, $/yr

PBP = pay back period

Findings

Radiant heaters are economical and practical for use in large open bay buildings and generally surpass convective forced-air heaters in heating large open bays by:

- providing increased thermal comfort at the floor level while reducing energy consumption

- being able to heat objects to just above the dew point temperature to prevent condensation and corrosion

- allowing heating flexibility with zone or whole building heating

ENERGY MONITORING AND CONTROL SYSTEMS (EMCS)

Energy monitoring and control systems (EMCS) can achieve significant energy savings. These systems are complex and require engineering studies before being installed in hangars. The following information aids in the preparation or the evaluation of engineering studies related to EMCS and hangars (see Ref 19).

Description

EMCS are supervisory control systems that interface with existing HVAC control loops to provide better control of HVAC systems. These systems can range from time clocks, implementing scheduled start/stop of equipment, to large scale systems capable of optimizing energy usage base-wide. An add-on to an existing large scale EMCS is probably the first preference, followed by single building controllers. The most common control strategies implemented at government installations with EMCS are described below.

1. Scheduled Start/Stop is the simplest of all control strategies to implement. The equipment is turned on and off at preselected times, which are selected to insure that the building comfort conditions are met during hours of building occupancy.

2. Optimized Start/Stop turns on equipment at the latest possible time and turns off the equipment at the earliest possible time while still meeting the building occupancy requirements. This strategy works well on HVAC equipment during the fall and spring or in mild climates. During times of worst-case outside temperature conditions, optimized start and stop would turn the equipment on and off at the same time the scheduled start and stop strategy would.

3. Duty Cycling turns off equipment for predetermined periods of time during part or all of the day. HVAC equipment is usually designed and installed for worst-case outside environmental conditions; these conditions seldom happen. Therefore, the HVAC equipment has the excess capacity to overcome the additional load placed on it by the duty-cycling program during normal conditions. The amount of time the HVAC

equipment can be shut down and still keep the building conditions within comfort limits depends on how much excess capacity the HVAC equipment has, what the normal outside environmental conditions are, and the required inside conditions. A typical duty-cycling scheme would turn the equipment off for 15 minutes out of an hour. Splitting the HVAC loads into roughly equal numbers and assigning them to different blocks of off times can result in significant dollar savings on electrical demand charges.

4. Day/Night Setback is used when HVAC equipment cannot be turned off, but the temperature required in the building can be raised or lowered, depending on whether the building is being cooled or heated. In most cases, every degree that is raised or lowered can save a tremendous amount of energy. A typical heating setback is from 65 to 55°F; a cooling setback (or setup as it is sometimes known) could be from 78 to 85°F. On large buildings that have both a heating and cooling requirement at the same time, care should be taken to insure that this control strategy does not consume more energy than letting the system operate at its normal temperature setpoints.

5. Ventilation/Recirculation controls the amount of outside air introduced to the building during nonoccupied hours. The outside air dampers are closed when the introduction of outside air would increase the load on the HVAC equipment and opened when the introduction of the outside air would decrease the load on the HVAC equipment. This strategy can be used in buildings which must maintain closed environmental conditions because of equipment operating requirements but are not occupied on a 24-hour-a-day basis.

6. Temperature Reset causes the heating system to lower the temperature of the supplied air or water and the cooling system to raise the temperature of the supplied chilled water or air. The temperature is reset until the area requiring the most heating and the area requiring the most cooling are just barely being satisfied. This control strategy works well on hot deck/cold deck, terminal reheat, hot water, chilled water, and cooling tower condenser water systems.

Feasibility Requirements

EMCS is generally feasible for all buildings with significant energy consumption from HVAC but not occupied 24 hr/day, 7 days/wk.

Energy Survey Data Requirements

To determine whether or not an EMCS is feasible for a particular situation, the designer should:

1. Determine design requirements of HVAC and building from as-built drawing
2. Perform a site survey and gather name plate data from equipment and general condition of equipment in the building
3. Obtain actual user requirements

Procedure For Calculating Feasibility And Energy Savings

To determine which control strategies are feasible and to calculate the annual energy savings, manual methods presented in the discussion of each type of strategy will give a reasonable degree of accuracy. For greater accuracy, computer methods should be used.

When calculating energy savings for systems on which more than one EMCS function may be applied, care must be taken not to duplicate savings or to calculate the same heating or cooling savings for both the secondary system and primary system serving it. For example, both an air handler and the chiller providing chilled water to the air conditioning/heating unit coil may be considered for Scheduled Start/Stop. The cooling savings for the space being served may be calculated in the savings analysis for the air handler or the chiller, but not both.

The timed-event programs - Scheduled Start/Stop, Day/Night Setback, Ventilation/Recirculation, and Optimum Start/Stop --are closely related, and the savings attributable to each is dependent on how the function is defined. An attempt has been made in the development of standard methods of determining energy savings to differentiate among these programs based on the descriptions found in Section II of Reference 20.

Scheduled Start/Stop may be applied to systems which can be shut down during unoccupied hours, such as chillers and air handlers serving noncritical areas. Day/Night Setback is to be applied to systems which cannot be completely shut down during unoccupied hours but can have thermostat set points set back. Optimum Start/Stop calculations are applicable only in conjunction with Scheduled Start/Stop for systems having auxiliary pumps or fans. Some heating and cooling energy may be saved by Optimum Start/Stop applied to night setback scheduling. Estimation of these savings would be difficult, however; therefore, only auxiliary savings are considered. The Ventilation/Recirculation program is applicable in conjunction with Scheduled Start/Stop or Day/Night Setback for air handlers which have been or are to be retrofitted with outside air damper control.

Standard methods for calculating yearly savings from each energy conservation strategy, as they apply to individual systems, have been developed and are discussed in this section. Computer methods are recommended for better accuracy when a building energy simulation computer program is available.

Each equation in the following sections results in an answer with units of energy per year. In most cases, cooling savings will be in kilowatt-hours per year, except where an absorption or steam-turbine-driven chiller is in operation. In that case, cooling savings will be in pounds of steam per year and needs to be converted to primary fuel source units for the on-site boiler, taking boiler efficiency into consideration. Answers for heating savings calculations will result in units of fuel consumption per year. The units could be cubic feet of natural gas per year or gallons of fuel oil per year or any other primary source of heat at the facility.

Scheduled Start/Stop.

1. Manual Method. The following savings calculations for HVAC equipment assume a low temperature override to system shutdown. If no low temperature limit is desired, then the average winter temperature (AWT) is used in place of the low temperature limit (LTL) and percent runtime (PRT) is equal to zero.

$$\text{Cooling savings} = [\text{BTT} \times \text{AZ} \times (\text{AST} - \text{SPP}) \times (168\ \text{hr/wk} - \text{HC}) \times \text{WKS} \times \text{CPT} \times \text{F}]/(12{,}000\ \text{Btu/ton-hr}) \quad (38)$$

$$\text{Heating savings} = [\text{BTT} \times \text{AZ} \times (\text{WSP} - \text{LTL}) \times (168\ \text{hr/wk} - \text{HC}) \times \text{WKW} \times \text{F}]/(\eta \times \text{HV}) \quad (39)$$

IF AWT > LTL, AWT rather than LTL is used.

$$\text{Ventilation cooling savings} = [\text{CFM} \times \text{POA} \times (4.5\ \text{lb/cfm-hr}) \times (\text{OAH} - \text{RAH}) \times (168\ \text{hr/wk} - \text{HC}) \times \text{WKS} \times \text{CPT} \times \text{F}]/(12{,}000\ \text{Btu/ton-hr}) \quad (40)$$

$$\text{Ventilation heating savings} = [\text{CFM} \times \text{POA} \times (1.08\ \text{Btu/cfm}^{\circ}\text{F-hr}) \times (\text{WSP} - \text{AWT}) \times (168\ \text{hr/wk} - \text{HC}) \times \text{WKW} \times \text{F}]/(\eta \times \text{HV}) \quad . . . . (41)$$

$$\text{Auxiliary savings} = \text{HP} \times \text{L} \times (0.746\ \text{kW/hp}) \times (168\ \text{hr/wk} - \text{H}) \times [\text{WKS} + (\text{WKW} \times (1\text{-PRT})] \times \text{F} \quad (42)$$

where:

AST = average summer temperature, °F

AWT = average winter temperature, °F

AZ = area of zone being served, ft^2

BTT = building thermal transmission, Btu/hr°F-ft^2

CFM = air handling capacity, ft^3/min

CPT = energy consumption per ton of refrigeration, kW/ton or lb/ton-hr

F = fraction of savings attributable to EMCS

H = hours of operation per week (use present time clock schedule or occupied hours plus 2 hours each morning)

HP = horsepower indicated on motor nameplate (total of continuously running fans and pumps)

HV = heating value of fuel, Btu/gal, Btu/kWh, etc.

L = load factor

LTL = low temperature limit, °F (usually 50 or 55°F)

OAH = average outside air enthalpy, Btu/lb

POA = present percent minimum outside air expressed as a decimal

PRT = percent running time during heating season shutdown period required to maintain a low limit temperature of 55°F expressed as a decimal. Use PRT = 0 if no low temperature limit is planned

RAH = return air enthalpy during normal operating hours. Use 29.91 Btu/lb for 78°F and 50% humidity. For other conditions, obtain values from a psychrometric chart

SSP = summer thermostat setpoint, °F

WKS = length of summer cooling season, wk/yr

WKW = length of winter heating season, wk/yr

WSP = winter thermostat setpoint, °F

η = overall heating system efficiency

2. Computer Method. Building loads and system operation are simulated, using a computerized energy analysis program. In the initial run it is assumed that the systems run 24 hr/day, 7 day/wk. In the second run, it is assumed that systems run only during occupied hours, plus 2 hours in the morning for warm-up or cool-down. Desired low limit temperatures are included when applicable. Fan kilowatts in computer runs are not included so that the difference in results is representative only of heating and cooling energy reduction. This heating and cooling energy savings can then be proportioned on a per-square-foot basis to other similar systems serving zones with similar building loads.

Cooling savings = Difference in electrical consumption of computer analysis runs

Heating savings = Difference in heating consumption of computer analysis runs

For auxiliary savings, the manual method should be consulted.

The following procedure determines the yearly savings from Scheduled Start/Stop of a domestic hot water (DHW) heater.

1. Calculate tank volume and surface area:

$$V = 0.785 \times DIAM^2 \times HT \quad (43)$$

$$A = (1.571 \times DIAM^2) + (3.14 \times D \times HT) \quad (44)$$

2. Use Figure 50 to determine the quantity:

$$E = \frac{Tw - Ts}{To - Ts} \quad (45)$$

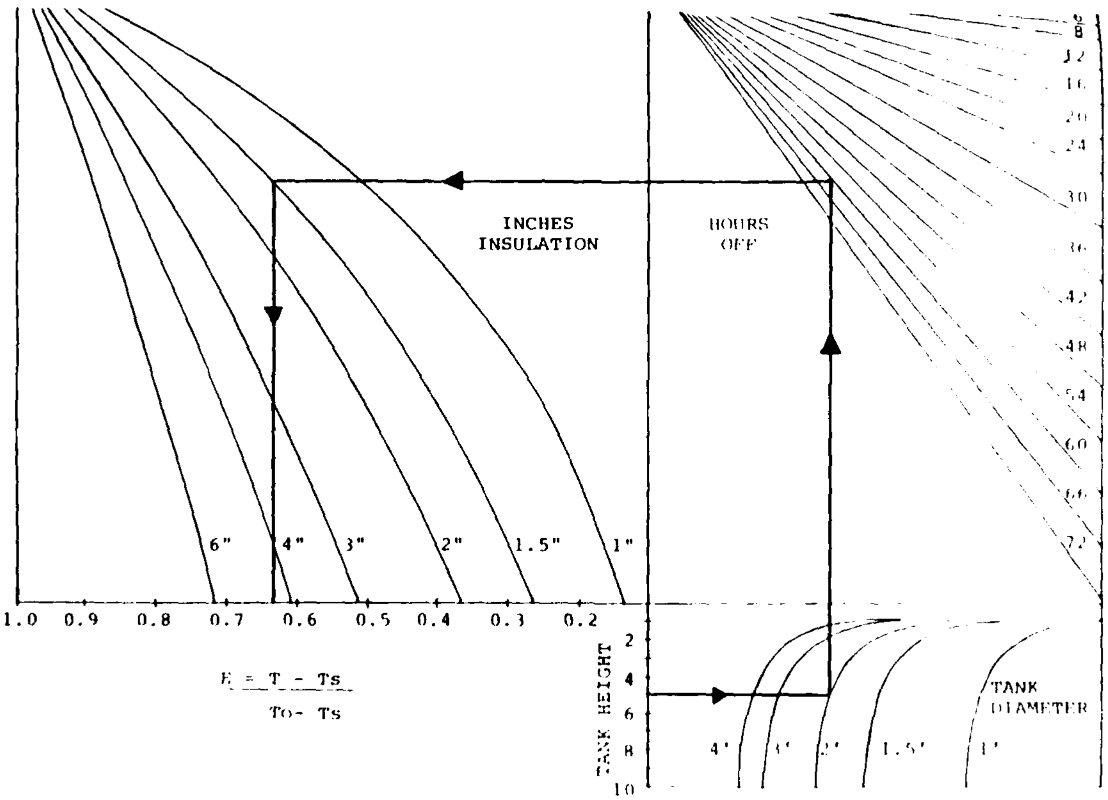


Figure 50. DHW off-time temperature drop.

3. Calculate the energy savings:

DHW heating savings =

$$\{[A \times (To-Ts) \times LSD \times (0.285\ Btu\text{-}in/ft^2hr^\circ F/INS)] - [V \times 62.4\ Btu/ft^{2\circ}F \times (To-Ts) \times (1-E)]\} \times NSD \times F/(\eta \times HV) \quad (46)$$

4. Repeat steps 2 and 3 for each different length of shutdown period and then total the savings.

where: A = surface area of tank, ft^2

DIAM = diameter of tank, ft

E = parameter determined from Figure 50

HT = height of tank, ft

INS = thickness of insulation, in.

LSD = length of shutdown period, hr

NSD = number of shutdown periods per year of a given length

Tw = water temperature at end of shutdown period, °F

To = hot water temperature setpoint, °F

Ts = average temperature of surroundings, °F

V = volume of tank, ft^3

If the system is currently started and stopped manually by a time switch, full credit cannot be taken for the above savings for the EMCS. Determining what savings may be attributed to the EMCS becomes a function of the reliability of the time switch system. Time switches can be effective devices for the reduction of energy consumption; however, they have several disadvantages. They do not take into account holiday operation, seasonal changes, or daily weather variations. They are also easily tampered with, bypassed, or overridden. The pins which activate actions may slide, thus causing system operation and energy consumption at unnecessary times. They must be checked often to ensure proper operation and must be reset manually every time a power outage occurs for any appreciable time period. Manual operation is subject to human error and forgetfulness.

The EMCS is capable of performing the same operations but without most of the difficulties described; it is not within the reach of tampering, and system operations are monitored constantly by the console operator. Therefore, the EMCS should be credited with some portion of these savings due to the increased reliability and the EMCS's ability to adjust and optimize start and stop times.

The fraction of savings, F, attributable to the EMCS shall be used to account for present timeclock or manual operation and future use of extended service capability of the system. Let F equal 1.0 if the system is presently operating around the clock, and no extended service is projected. Otherwise, the value shall be between 0 and 1.0, depending on extension of operation and the reliability of the present control as determined during the field survey.

Duty Cycling. This function is applicable to electrical loads under 30-hp nameplate rating; however, the savings calculations apply only to constant loads. Duty cycling of loads which already cycle under local controls may save energy by essentially overriding the local thermostat setting, but these savings would be difficult to estimate and so are not included in the analysis. For motors above 30 hp, the savings are offset by added maintenance cost due to excessive wear on belts and bearings caused by frequent cycling.

1. Manual Method. Assume the system may be shut down for an average of 10 min/hr. The savings resulting from this function are fan or other auxiliary energy and outside air heating and cooling energy. Outside air loads are difficult to determine since they actually depend on space load conditions. If there is a net cooling load in the space and if the outside air is below 75°F, the outside air actually reduces energy consumption, which is often the case in commercial buildings during the heating season. Therefore, ventilation savings will not be credited by the manual method.

Auxiliary savings =

$$HP \times L \times 10/60 \times (0.746\ kW/HP) \times HW \times (52\ wk/yr) \qquad (47)$$

where HW is equal to hours of operation per week (use number of hours of occupancy assuming duty cycling is not desirable during warmup). Fraction of time the system is shut down assumes 10 minutes out of each hour (10/60).

2. Computer Method. Building loads and systems operation are simulated by using a computerized energy analysis program capable of calculating annual energy consumption. In the initial run, the system is scheduled to run during occupied hours plus 2 hours in the morning. On the second run, the system is scheduled to run for only 50 minutes of each hour except the first 2. It is important to use accurate actual ventilation air quantities as input to the program, if possible. Dry bulb or enthalpy economizer is included in both runs if either exists or is to be implemented for the system by the EMCS. Fan kilowatt input are not included in the computer runs so that the difference in results only represents heating and cooling energy reductions.

Cooling savings = Difference in electrical consumption of computer analysis runs

Heating savings = Difference in heating consumption of computer analysis runs

For Auxiliary savings, the manual method should be consulted.

Optimum Start/Stop. Auxiliary savings from this function are derived by minimizing the necessary warm-up or cool-down time prior to occupancy and by shutdown of the system as early as possible before the end of the occupancy period. Early shutdown is applicable only where ventilation is not critical and most of the occupants vacate the building at the scheduled time. Cooling and heating savings obtainable by keeping OA dampers closed during warm-up/cool-down times are accounted for in the Ventilation and Recirculation savings calculations. While a small amount of energy may be saved due to less run time of cycling loads (cooling tower fans or unit heaters), it is difficult to estimate and is not included in this analysis.

Warm-up auxiliary savings =

$$HP \times L \times (0.746 \text{ kW/hp}) \times [(WH \times AND) - ERT)] \times (DAY/7 \text{ day/wk}) \quad \ldots\ldots\ldots \quad (48)$$

Cool-down auxiliary savings* =

$$HP \times L \times (0.746 \text{ kW/hp}) \times (CH - 0.75 \text{ hr/day}) \times (365 \text{ day/yr} - AND) \times (DAY/7 \text{ day/wk}) \quad \ldots\ldots\ldots \quad (49)$$

where: AND = annual number of days total that warm-up is required, day/yr

CH = present cool-down time before occupancy, hr/day. Use either the actual time presently scheduled for cool-down by an existing timeclock or 2 hours to correspond to Scheduled Start/Stop savings calculations

DAY = equipment operation, day/wk

ERT = total equipment running time required for warm-up, hr/yr

WH = present warm-up time before occupancy, hr/day. Use either the actual time presently scheduled for warm-up by an existing timeclock or 2 hours to correspond to Scheduled Start/Stop savings calculations

Ventilation and Recirculation. Savings from this function are a result of control of OA dampers. All calculations assume that a 15-minute purge of ventilation air is necessary prior to occupancy.

*This calculation assumes a 45-minute (0.75 hour) cool-down time is required per day during the days of the year not requiring warm-up. This is a conservative estimate; in most parts of the country, a 15-minute purge would probably be sufficient in mild weather.

The following calculation is applicable to systems which are shut down by the Scheduled Start/Stop function and is restricted to the period of time during warm-up or cool-down prior to occupancy. No cool-down ventilation savings is included in the analysis based on the assumption that early morning outside air adds a negligible amount to the cooling load and in fact may lessen the load through an economizer effect.

Warm-up ventilation heating savings =

$$[CFM \times POA \times (WSP - AWT) \times (1.08\ Btu/cfm^{\circ}F\text{-}hr) \times AND \times (WH - 0.25\ hr/day)]/(\eta \times HV) \quad (50)$$

The next two calculations are applicable to fan systems which must maintain environmental conditions but may eliminate outside air during periods when the building is unoccupied.

Ventilation cooling savings =

$$\{CFM \times POA \times (4.5\ lb/cfm\text{-}hr) \times (OAH\text{-}RAH) \times (UH - [(0.25\ hr/day \times DAY)] \times WKS \times CPT]\}/12{,}000\ Btu/ton\text{-}hr \quad (51)$$

Ventilating heating savings =

$$\{CFM \times POA \times (1.08\ Btu/cfm^{\circ}F\text{-}hr) \times (WSP - AWT) \times [(UH\text{-}(0.25\ hr/day \times DAY)] \times WKW\}/(\eta \times HV) \quad (52)$$

where UH is unoccupied building time in hours per week.

Day/Night Setback. This strategy would be applied, instead of Scheduled Start/Stop, to systems with no auxiliaries such as steam radiation. It is also applicable to systems which serve critical areas with temperature, humidity, or pressure requirements that will allow a small setpoint adjustment. However, the system cannot be stopped altogether. If outside air, OA, dampers can be closed during the setback period, ventilation savings are possible and should be calculated under the Ventilation and Recirculation strategy. Only a manual method can be used.

Cooling savings =

$$BTT \times AZ \times SU \times (168 - HN) \times WKS \times CPT/(12{,}000\ Btu/ton\text{-}hr) \quad \ldots\ldots\ldots (53)$$

Heating savings =

$$BTT \times AZ \times SD \times (168 - H) \times WKW/(\eta \times HV) \quad (54)$$

where: HN = hours of operation per week during which the normal setpoint applies

SD = thermostat setdown for unoccupied periods during the heating season, °F

SU = thermostat setup for unoccupied periods during the cooling season, °F

Reheat Coil Reset.

1. Manual Method. A computer simulation is recommended for these savings calculations and is required for accurately determining the savings from Reheat Coil Reset, when economizer control is also applied to the system. The cooling savings with an economizer will be one-third to four-fifths of the savings without an economizer due to the reduction of mechanical cooling already obtained by the economizer control.

Cooling savings (no economizer)* =

$$[HH \times CFM \times (4.5 \text{ min lb/hr ft}^3) \times WKS \times RHR \times (0.6 \text{ Btu/lb}) \times CPT]/12{,}000 \text{ Btu/ton-hr} \qquad (55)$$

Heating savings** =

$$[HH \times CFM \times (1.08 \text{ Btu/cfm-hr}^\circ F) \times (52 \text{ wk/yr}) \times RHR]/(\eta \times HV) \qquad (56)$$

where: HH = hours of operation per week (hours of occupancy plus 1 hour for each day)

RHR = Reheat system cooling coil discharge reset, °F (up to 5 or 6°F is possible, depending on the system). If a better estimate of possible reset is not available, use 3°F.

2. Computer Method. Simulate building loads and system operation with a computerized energy analysis program. Preferably the program used should have simulation routines for selecting the zones with the greatest cooling demand and calculating the necessary cooling coil leaving air temperature or at least the capability of a reset schedule. In order to approximate the savings from this function, run the program

*This equation assumes a 1°F cooling coil temperature increase is equivalent to a 0.6 Btu/lb change in enthalpy.

**To account for holiday shutdown or for a system that does not operate year-round, the 52 wk/yr term can be adjusted accordingly.

once using a constant cooling coil setpoint temperature; then run a second time, simulating variable reset based on a discriminator scheme or a reset schedule. Be sure to include economizer control when applicable.

Cooling savings = Difference in electrical consumption of computer analysis runs

Heating savings = Difference in heating consumption of computer analysis runs

Hot Deck/Cold Deck Temperature Reset.

1. Manual Method. A computer simulation is recommended for these savings calculations and is required for accurately determining the savings from Hot Deck/Cold Deck Temperature Reset when economizer control is also applied to the system. Because of the reduction of mechanical cooling already obtained by the economizer control, the cooling savings with an economizer can be as little as one-fifth of the savings without an economizer.

Cooling savings (no economizer)* =

$$[HH \times CFM \times CD \times (4.5 \text{ min lb/hr ft}^3) \times WKS \times SCDR \times (0.6 \text{ Btu/lb}) \times CPT]/12{,}000 \text{ Btu/ton-hr} \qquad (57)$$

Heating savings =

$$[HR \times CFM \times HD \times (1.08 \text{ min Btu/hr ft}^3{}^\circ F) \times (WKS \times SHDR + WKW \times WHDR)]/(\eta \times HV) \qquad (58)$$

where: CD = Fraction of total air passing through the cold deck (assume 0.50 if no other information is available)

HD = Fraction of total air passing through the hot deck (assume 0.50 if no other information is available)

HR = Required number of hours of operation per week (assume hours of occupancy plus 1 hr/day)

SCDR = Summer cold deck reset, °F. (The average reset is a function of the system. If an estimate is not available, use 2°F.)

*This equation assumes a 1°F cold deck temperature increase is equivalent to a 0.6 Btu/lb change in enthalpy.

SHDR = Summer hot deck reset, °F. (The average reset that will result from this function is dependent on the air handler capacity relative to the loads in the space it serves. If an estimate of the possible reset is not available, use 3°F.)

WHDR = Winter hot deck reset, °F. (Again, the average reset is a function of the system. If an estimate is not available, use 2°F.)

2. Computer Method. Simulate building loads and system operation with a computerized energy analysis program. The program used should have simulation routines necessary to select the zones with the greatest heating and cooling demands and then calculate the necessary hot and cold deck leaving temperatures. In order to approximate the savings from this function, run the program once, using constant deck setpoint temperatures; then run a second time, simulating variable deck temperatures based on a discriminator control scheme. Be sure to include economizer control when applicable.

Cooling savings = Difference in electrical consumption of computer analysis runs

Heating savings = Difference in heating consumption of computer analysis runs

Hot Water Outside Air Reset. Boiler temperature reset saves energy by reducing heat losses through the heating system and flue gases and by providing more exact control at the end use point. This last item provides savings by reducing overheating of spaces at less than maximum loads because of control valve insensitivity in those operating ranges. Reset of hot water supply temperature from a converter produces savings similarly. No exact means of quantifying these savings is known; however, experience indicates these savings should be a function of the annual equivalent full load hours of system operation and the total capacity of the system.

$$\text{Heating savings} = (HFLH \times EI \times CAP)/\eta \times HV \qquad (59)$$

where: CAP = maximum capacity of device(s), Btu/hr

EI = Efficiency increase expressed as a decimal (use 0.01 if no better estimate is available)

HFLH = annual equivalent full load hours for heating, hr/yr

Chiller Water Temperature Reset. Reset of chilled water supply temperatures results in energy savings due to the increased efficiency of the refrigeration machine. Check to be sure that a chilled water

controller may be applied to the particular manufacturer's chiller being considered. The savings will vary, depending on the machine, the amount of reset, and the load on the equipment. The amount of reset generally ranges between 2 and 5°F, so a conservative estimate of 2°F was used in the calculation.

$$\text{Cooling savings} = \text{TON} \times \text{CPT} \times \text{CFLH} \times 2^{\circ}\text{F} \times \text{REI} \tag{60}$$

where: CFLH = equivalent full-load hours for cooling, hr/yr

REI = rate of efficiency increase per °F increase of chilled water temperature. Use the following values for the type of machine used.

Type of Machine	REI Value
Screw compressor machine	0.024/°F
Centrifugal (electric or machine turbine)	0.017/°F
Reciprocal machine	0.012/°F
Absorption machine	0.006/°F

TON = Chiller capacity in tons. If chiller capacity is not available and nameplate electrical data on the chiller motor is, use the full-load kilowatt input in place of (TON x CPT).

Condenser Water Temperature Reset. Decreasing the condenser water temperature also increases the efficiency of chillers, but care must be taken not to exceed the equipment limitations, particularly in absorption machines. The implementation of condenser water reset may result in greater fan energy consumption. If a cooling tower fan cycles on and off, the on time will be increased, consuming more auxiliary energy. If it runs continuously with valve bypass control to maintain constant entering condenser water temperature and can be cycled when the EMCS function is applied, then additional auxiliary energy can be saved. An adjustment to account for these conditions has been included in the savings analysis.

The calculation procedures requires four steps:

1. Calculate the average reduction in condenser water temperature which is achievable:

$$\text{RCWT} = \text{PCWT} - \text{ACWT} \tag{61}$$

where: ACWT = average condenser water temperature possible, °F

PCWT = present condenser water temperature, °F (usually set at 85°F)

RCWT = reduction in condenser water temperature which is achievable, °F

2. Use Figure 51 to determine the percent efficiency increase (PEI) of the chiller based on RCWT from above.

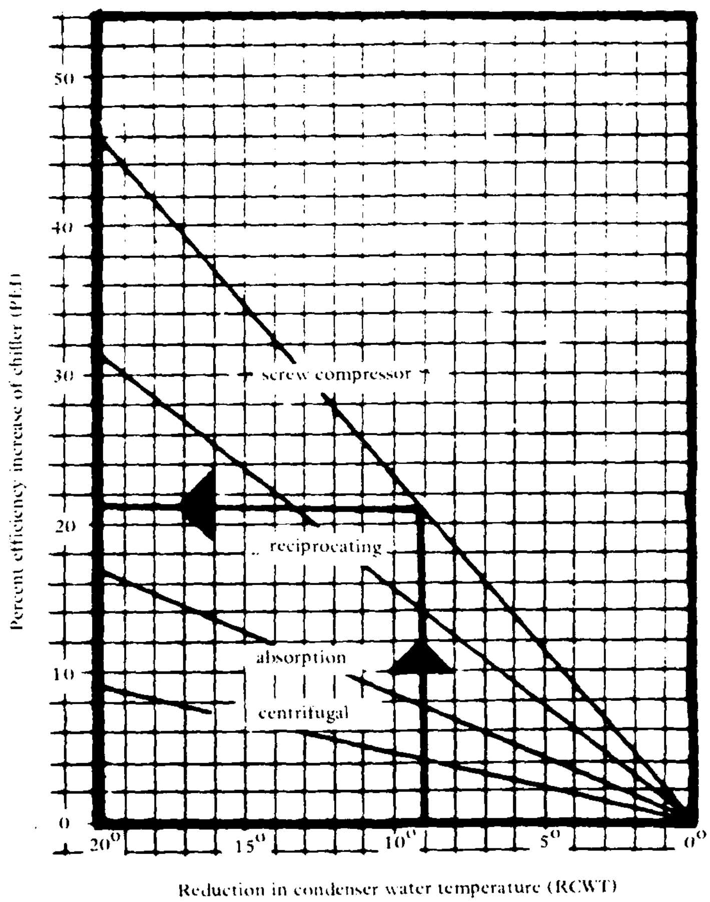


Figure 51. Curve for determining percent efficiency increase.

3. Determine the adjusted efficiency increase (AEI) of the chiller:

If fan runs continuously, but will be cycled,

$$AEI = \frac{PEI + 5.5}{100} \tag{62}$$

If fan cycles,

$$AEI = \frac{PEI - 2.8}{100} \tag{63}$$

where: AEI = adjusted efficiency increase of the chiller due to condenser water reset

PEI = percent efficiency increase of the chiller

4. Calculate the cooling savings:

$$\text{Cooling savings} = \text{TON} \times \text{CPT} \times \text{CFLH} \times \text{AEI} \qquad (64)$$

where CFLH is equal to equivalent full load hours for cooling in hours/year.

Lighting Control. This function is applicable to relay-operated zoned lighting. The following calculation is for one zone of lighting.

$$\text{Electrical savings} = \text{kW} \times (168\ \text{hr/wk-HO})) \times 52\ \text{wk/yr} \times F \qquad (65)$$

where: F = fraction of savings attributable to EMCS*

HO = hours of operation per week (use hours of occupancy)

kW = total kilowatt consumption of lights in the zone

Installation Considerations

1. Life expectancy of the EMCS is 15 years.

2. Installation cost is site specific.

3. Using either a microprocessor-based single building controller or an add-on to existing large-scale EMCS is strongly recommended instead of using time clocks or other low cost devices (see Table 16 for controller characteristics).

ANNUAL HANGAR ENERGY INSPECTION

Hangar inspection should be conducted systematically during late spring or early summer. Seven areas must be inspected for discrepancies that cause excessive heat loss. These are listed, along with the location and physical characteristics of all discrepancies that should be recorded. Inspection guidelines with suggested corrective measures for discrepancies are also listed.

*This factor is a subjective measure of how diligently the lights are turned off manually at the present.

Table 16. Characteristics of Typical Building Controllers

Single Building Controllers	Energy Conservation Control Characteristics						Range of Prices ($)	Comments
	Scheduled Start/Stop	Optimized Start/Stop	Duty Cycling	Day/Night Setback	Ventilation/Recirculation	Temperature Reset		
Time Clocks	yes			optional			25-1,600	Can have significant maintenance problems
Duty Cyclers	yes		yes				150-20,000	Can save on electrical demand charges
Specialized Equipment	yes	yes		yes	yes	yes	200-20,000	A specific controller can only be used on the particular type of equipment it was designed for
Programmable Controllers	yes	yes	yes	yes	yes	yes	300-30,000	Replacement for relay racks; operates in harsh environments; works best when it is controlling based on on/off conditions
Micro EMCS	yes	yes	yes	yes	yes	yes	750-30,000	Works well in single buildings where control strategies implemented do not need to be changed frequently
Small EMCS	yes	yes	yes	yes	yes	yes	1,200 and up	Operator interface so building O&M staff may monitor, override, and change control strategies
Large EMCS	yes	yes	yes	yes	yes	yes	5,000 and up	Centralized operator interface and basewide control strategies may be implemented

1. Aircraft access doors
   (a) Door seal condition:
       good
       bad (repair or replace)
   (b) Door seal material:
       nylon brush
       rubber (replace with nylon brush seals)
   (c) Door alignment:
       good
       bad (realign)
   (d) Insulation:
       yes (___ inches, material)
       no (insulate)
   (e) Surface condition:
       good
       bad (repair, seal holes, etc.)
   (f) Ease of opening:
       easy
       difficult (lubricate or repair)

2. Vehicle access doors
   (a) Vehicle access doors installed:
       yes
       no (install)
   (b) Door seal condition:
       good
       bad (repair or replace)
   (c) Surface condition:
       good
       bad (repair, seal holes, etc.)
   (d) Insulation:
       yes (___ inches, material)
       no (insulate)
   (e) Alignment:
       good
       bad (realign)
   (f) Flexible vinyl strip doors installed:
       yes
       no (install)
   (g) Door operation:
       easy
       difficult (repair or replace)

3. Personnel access doors
   (a) Door seal condition:
       good
       bad (repair or replace)
   (b) Surface condition:
       good
       bad (repair, seal holes, etc.)

3. (continued)
   (c) Door function properly:
       yes
       no (repair or replace)
   (d) Door frequently used:
       no
       yes (install flexible vinyl strip door or entrance vestibule)

4. Exterior walls
   (a) Insulation:
       yes (___ inches, material)
       no (insulate)
   (b) Surface condition:
       good
       bad (repair, seal holes, etc.)
   (c) Floor/wall caulking:
       yes
       no (caulk)
   (d) Ceiling/wall caulking:
       yes
       no (caulk)

5. Roof (a) Insulation:
       yes (___ inches, material)
       no (insulate)
   (b) Surface condition:
       good
       bad (repair, seal holes, etc.)
   (c) Air vents closeable:
       yes
       no (repair or install closeable vents)

6. Windows
   (a) Condition:
       good
       bad (replace broken, cracked, or missing panes)
   (b) Caulking:
       good
       bad (caulk)
   (c) Daytime electric lighting:
       no
       yes (reduce window area by replacement with wall/door structural material or cover with insulation)
   (d) Double glazing or storm windows:
       yes
       no (install double glazing or storm windows)
   (e) Openable windows close and seal:
       yes
       no (repair, replace seals, etc.)

7. Heating system

(a) Aircraft door deactivation switch:

- yes
- no (install deactivation switch)

(b) Temperature setback:

- yes
- no (install for off-shift periods)

(c) Thermostat calibration:

- good
- bad (recalibrate)

(d) Steam traps:

- good
- bad (replace)

(e) Unit heaters operable:

- yes
- no (repair or replace)

(f) Thermostat operable:

- yes
- no (repair or replace)

(g) Floor level air temperature:

- ___°F (set thermostat to 65°F or less, dependent upon activity in hangar; higher temperatures are required for some painting and corrosion control functions)

(h) Energy monitoring and control system installed:

- yes
- no (install, if feasible)

CONCLUSIONS AND RECOMMENDATIONS

The two major sources of heating-related energy loss in hangars are air infiltration and stratification. Reduction in air infiltration losses can be obtained by:

- using nylon brush door seals
- maintaining the structural integrity of the hangar's exterior surfaces, doors and windows
- sealing cracks between hangar walls and its roof and floor
- reducing stratification

Stratification increases hangar thermal losses by increasing air infiltration ratio and by increasing heat losses through roof and upper wall surfaces. Stratification can be reduced by maintaining thermostat settings at or below 60°F, by installing a cold air jet destratification system, or by installing a radiant heating system.

Possible solutions for reducing stratification and its consequent heat loss and thus conserving energy consumption are:

1. <u>Radiant Heating</u>

Radiant heating is ideal for large open structures such as hangars. Unlike convection heating, radiant heating heats personnel and objects directly and interior air indirectly. Thus, comfort levels for personnel can be achieved and the amount of energy required for space heating reduced though internal air temperatures are lower. At the same time, the lower internal air temperatures reduce the severity of stratification and air infiltration losses.

2. <u>Energy Monitoring and Control System (EMCS)</u>

The control of hangar heating, lighting, and other electromechanical devices by an EMCS can save significant amounts of energy. EMCS should be considered for all hangars, but especially for hangars utilized less than 12 hr/day.

3. <u>Hangar Energy Inspection</u>

An annual hangar inspection can help ensure that heating-related energy consumption levels are lowered. The purpose of the inspection, which should be done during late spring or early summer, is to provide

timely identification and documentation of discrepancies which contribute to excessive heat losses. This will provide Public Works personnel with an opportunity to correct discrepancies prior to the heating season.

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LIST OF SYMBOLS

A	Surface area of tank, ft^2
a	Surface area of the emitter, ft^2
ACWT	Average condenser water temperature possible, ^oF
A_d	Door area, ft^2
AEI	Adjusted efficiency increase of the chiller due to condenser water reset
AFCU	Fuel units (i.e., gallons, Btu)
AHC	Annual heating costs, $
AND	Annual number of days total that warmup is required, day/yr
A_o	Total crack area, ft^2
AST	Average summer temperature, ^oF
A_w	Total vertical surface area
AWT	Average winter temperature, ^oF
AZ	Area of zone being served, ft^2
BTT	Building thermal transmission, Btu/hr^oF-ft^2
C	Shielding coefficient
CAP	Maximum capacity of devices, Btu/hr
CD	Fraction of total air passing through the cold deck (assume 0.50, if no other information is available)
CFLH	Equivalent full-load hours for cooling, hr/yr
CFM	Air handling capacity, ft^3/min
CH	Present cool-down time before occupany, hr/day

c_p	Coefficient of heat for air, Btu/lb-°F
c_p	Specific heat of air ≅ 0.24 Btu/lb-°F
CPT	Energy consumption per ton of refrigeration, kW/ton or lb/ton-hr
d	Degree-day
DAY	Equipment operation, day/wk
D	Degree-days
DHW	Domestic hot water
dia_i	Nozzle diameter, in.
DIA	Diameter of tank, ft
e	Hourly energy loss, Btu/hr
EI	Efficiency increase expressed as a decimal (use 0.01 if no better estimate is available)
ERT	Equipment run time total required for warmup, hr/yr
E_V	Energy saved, Btu
F	Fraction of savings attributable to EMCS
f_s	Stack effect parameter
f_w	Wind effect parameter
g	Acceleration of gravity, fpm^2
H	Interior height, ft
HC	Hours of operation per week using present time clock schedule or occupied hours plus 2 hours each morning
HD	Fraction of total air passing through the hot deck (assume 0.50 if no other information is available)
HH	Hours of operation per week assuming hours of occupancy plus 1 hr/day
H_L	Calculated total building heat loss, Btu/hr

HN	Hours of operation per week during which the normal set point applies
HO	Hours of operation per week using hours of occupancy
HP	Horsepower indicated on motor nameplate (total of continuously running fans and pumps)
HR	Required number of hours of operation per week assuming hours of occupancy plus 1 hr/day
HT	Height of tank, ft
HV	Heating value of fuel, Btu/gal, Btu/kWh, etc.
HW	Hours of operation per week using number of hours of occupancy and assuming duty cycling is not desirable during warmup
HFLH	Annual equivalent full load hours for heating, hr/yr
I	Air changes, no./hr
INS	Thickness of insulation, in.
K	Terrain coefficient
k	Stephan-Boltzman constant ($0.17^3 \times 10^{-8}$ Btu/hr-ft^2°R^4)
kW	Total kilowatts consumed by lights in the zone
L	Load factor
L_A	Air infiltration energy losses during a heating season
LSD	Length of shutdown period, hr
LTL	Low temperature limit, °F (usually 50 or 55°F)
N	Total number of days in a heating season
N_d	Number of destratifiers to be installed
NSD	Number of shutdown periods of a given length per year
OAH	Average outside air enthalpy, Btu/lb

p	Destratifier electric power consumption
PBP	Pay-back period
PCS	Project cost savings for fuel ($/hr)
PCWT	Present condenser water temperature (usually set at 85°F)
PEI	Percent efficiency increase of the chiller
POA	Present percent minimum outside air expressed as a decimal
PRT	Percent running time during heating season shutdown period required to maintain a low limit temperature of 55°F expressed as a decimal. (Use PRT = 0 if no low temperature limit is planned)
Q	Air infiltration, cfm
q	Rate of emission or emissive power, Btu/hr
Q_d	Destratifier air movement ft³/min (cfm)
Q_f	Destratifier flow or fan air movement, ft³/hr
q_f	Fan flows, cfm
Q_o	Open air leakage, lb/min
Q_r	Reduction in air leakage, lb/min
Q_v	Vinyl door leakage ratio, lb/min
R	Leakage from horizontal cracks
RAH	Return air enthalpy during normal operating hours (for 78°F and 50% humidity, use 29.91 Btu/lb; for other conditions, obtain values from a psychrometric chart)
RCWT	Reduction in condenser water temperature which is achievable, °F
REI	Rate of efficiency increase per degree-Fahrenheit increase of chilled water temperature
RHR	Reheat system cooling coil discharge reset, °F (up to 5 or 6°F is possible, depending on the system. If a better estimate of possible reset is not available, use 3°F.)

S	Wind speed, mph
$\bar{S}$	Average wind speed during heating season, mph
SCDR	Summer hot deck reset, °F (the average rest that will result from this function is dependent on the air handler capacity relative to the loads in thespace it serves. If an estimate of the possible reset is not available, use 3°F.)
SD	Thermostat setdown for unoccupied periods during the heating season, °F
SHDR	Summer hot deck reset, °F. (The average reset that will result from this function is dependent on the air handler capacity relative to the loads in the space it serves. If an estimate of the possible reset is not available, use 3°F.)
SSP	Summer thermostat setpoint, °F
SU	Thermostat setup for unoccupied periods during the cooling season, °F
T	Interior air temperature, °R
t	Open door duration, hr
T_A	Average outside temperature for the day, °F
T_a	Ambient air temperature, °F
T_c	Inside air temperature 1 foot below the ceiling, °F
T_d	Outside design temperature, °F
T_f	Inside air temperature 1 foot above the floor, °F
T_i	Inside design temperature, °F
To	Hot water temperature setpoint, °F
TON	Chiller capacity, tons (if chiller capacity is not available and nameplate electrical data on the chiller motor is, use the full-load kilowatt input in place of TON x CPT)
Ts	Average temperature of surroundings, °F
Tw	Water temperature at end of shutdown period, °F

UH	Unoccupied building time, hr/wk
U_r	Residual air velocity at distant W, fpm
U_V	Air exit velocity at destratifier nozzle, fpm
V	Volume (hangar, building, tanks, chamber), ft^3
W	Wind speed, fpm
W_f	Width of hangar, ft
WH	Present warmup time before occupancy, hr/day (use either the actual time presently scheduled for warmup by an existing time-clock or 2 hours to correspond to Scheduled Start/Stop savings calculations)
WHDR	Winter hot deck reset, °F (again, the average reset is a function of the system; if an estimate is not available, use 2°F.)
WKS	Length of summer cooling season, wk/yr
WKW	Lenth of winter heating season, wk/yr
WSP	Winter thermostat setpoint, °F
X	Difference between the ceiling and floor leakage
α	Terrain coefficient
β	$f(T_f)$
γ	$f(T_f)$
ε	Emissivity of the emitter surface, dimensionless
ρ	Density of air, lb/ft^3
θ	Absolute temperature of the emitter surface, °R
η	Overall heating system efficiency, %/100
ΔP	Hangar's inside/outside pressure difference, psi
ΔQ	Differential leakage between seals, ft^3/min
ΔQ_y	Differential leakage, ft^3 per heating season
ΔT	Inside/outside air temperature difference, °F

ΔT_c	Ceiling/floor temperature difference, °F
ΔT_d	Inside/outside temperature difference at design temperature, °F
ΔT_A	Average inside/outside temperature difference for a hangar for a heating season
ΔT_S	Floor/ceiling temperature difference, stratified, °F
ΔT_D	Floor/ceiling temperature difference, destratified, °F

Appendix A

DESIGN TABLES, AIR INFILTRATION

DESIGN CONDITIONS: WIND= 0 MPH; TEMP.=-10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.34
500000	40	1.53
500000	45	1.72
500000	50	1.92
1000000	35	.95
1000000	40	1.08
1000000	45	1.22
1000000	50	1.35
1500000	35	.77
1500000	40	.88
1500000	45	.99
1500000	50	1.10
2000000	35	.67
2000000	40	.76
2000000	45	.86
2000000	50	.96
2500000	35	.60
2500000	40	.68
2500000	45	.77
2500000	50	.85
3000000	35	.54
3000000	40	.62
3000000	45	.70
3000000	50	.78
3500000	35	.50
3500000	40	.58
3500000	45	.65
3500000	50	.72
4000000	35	.47
4000000	40	.54
4000000	45	.61
4000000	50	.67

DESIGN CONDITIONS: WIND= Ø MPH; TEMP.= Ø DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
5ØØØØØ	35	1.24
5ØØØØØ	4Ø	1.42
5ØØØØØ	45	1.6Ø
5ØØØØØ	5Ø	1.77
1ØØØØØØ	35	.88
1ØØØØØØ	4Ø	1.ØØ
1ØØØØØØ	45	1.13
1ØØØØØØ	5Ø	1.25
15ØØØØØ	35	.71
15ØØØØØ	4Ø	.82
15ØØØØØ	45	.92
15ØØØØØ	5Ø	1.Ø2
2ØØØØØØ	35	.62
2ØØØØØØ	4Ø	.71
2ØØØØØØ	45	.8Ø
2ØØØØØØ	5Ø	.88
25ØØØØØ	35	.55
25ØØØØØ	4Ø	.63
25ØØØØØ	45	.71
25ØØØØØ	5Ø	.79
3ØØØØØØ	35	.5Ø
3ØØØØØØ	4Ø	.58
3ØØØØØØ	45	.65
3ØØØØØØ	5Ø	.72
35ØØØØØ	35	.47
35ØØØØØ	4Ø	.53
35ØØØØØ	45	.6Ø
35ØØØØØ	5Ø	.67
4ØØØØØØ	35	.44
4ØØØØØØ	4Ø	.5Ø
4ØØØØØØ	45	.56
4ØØØØØØ	5Ø	.62

DESIGN CONDITIONS: WIND= 0 MPH; TEMP.= 10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.13
500000	40	1.29
500000	45	1.46
500000	50	1.62
1000000	35	.80
1000000	40	.91
1000000	45	1.03
1000000	50	1.14
1500000	35	.65
1500000	40	.74
1500000	45	.84
1500000	50	.93
2000000	35	.56
2000000	40	.64
2000000	45	.73
2000000	50	.81
2500000	35	.50
2500000	40	.58
2500000	45	.65
2500000	50	.72
3000000	35	.46
3000000	40	.53
3000000	45	.59
3000000	50	.66
3500000	35	.42
3500000	40	.49
3500000	45	.55
3500000	50	.61
4000000	35	.40
4000000	40	.45
4000000	45	.51
4000000	50	.57

DESIGN CONDITIONS: WIND= Ø MPH; TEMP.= 20 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.01
500000	40	1.16
500000	45	1.30
500000	50	1.45
1000000	35	.71
1000000	40	.82
1000000	45	.92
1000000	50	1.02
1500000	35	.58
1500000	40	.67
1500000	45	.75
1500000	50	.83
2000000	35	.50
2000000	40	.58
2000000	45	.65
2000000	50	.72
2500000	35	.45
2500000	40	.51
2500000	45	.58
2500000	50	.64
3000000	35	.41
3000000	40	.47
3000000	45	.53
3000000	50	.59
3500000	35	.38
3500000	40	.43
3500000	45	.49
3500000	50	.54
4000000	35	.35
4000000	40	.41
4000000	45	.46
4000000	50	.51

DESIGN CONDITIONS: WIND= 0 MPH; TEMP.= 30 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	.88
500000	40	1.00
500000	45	1.13
500000	50	1.25
1000000	35	.62
1000000	40	.71
1000000	45	.80
1000000	50	.88
1500000	35	.50
1500000	40	.58
1500000	45	.65
1500000	50	.72
2000000	35	.44
2000000	40	.50
2000000	45	.56
2000000	50	.62
2500000	35	.39
2500000	40	.44
2500000	45	.50
2500000	50	.56
3000000	35	.35
3000000	40	.41
3000000	45	.46
3000000	50	.51
3500000	35	.33
3500000	40	.38
3500000	45	.42
3500000	50	.47
4000000	35	.31
4000000	40	.35
4000000	45	.40
4000000	50	.44

DESIGN CONDITIONS: WIND= 5 MPH: TEMP.=-10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.44
500000	40	1.63
500000	45	1.83
500000	50	2.03
1000000	35	1.01
1000000	40	1.15
1000000	45	1.29
1000000	50	1.43
1500000	35	.83
1500000	40	.94
1500000	45	1.06
1500000	50	1.17
2000000	35	.72
2000000	40	.81
2000000	45	.91
2000000	50	1.01
2500000	35	.64
2500000	40	.73
2500000	45	.82
2500000	50	.91
3000000	35	.58
3000000	40	.66
3000000	45	.75
3000000	50	.83
3500000	35	.54
3500000	40	.61
3500000	45	.69
3500000	50	.76
4000000	35	.50
4000000	40	.57
4000000	45	.64
4000000	50	.71

DESIGN CONDITIONS: WIND= 5 MPH; TEMP.= 0 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.34
500000	40	1.53
500000	45	1.71
500000	50	1.90
1000000	35	.95
1000000	40	1.08
1000000	45	1.21
1000000	50	1.34
1500000	35	.77
1500000	40	.88
1500000	45	.99
1500000	50	1.09
2000000	35	.67
2000000	40	.76
2000000	45	.85
2000000	50	.95
2500000	35	.60
2500000	40	.68
2500000	45	.76
2500000	50	.85
3000000	35	.55
3000000	40	.62
3000000	45	.70
3000000	50	.77
3500000	35	.50
3500000	40	.57
3500000	45	.64
3500000	50	.71
4000000	35	.47
4000000	40	.54
4000000	45	.60
4000000	50	.67

DESIGN CONDITIONS: WIND= 5 MPH; TEMP.= 10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.24
500000	40	1.41
500000	45	1.58
500000	50	1.75
1000000	35	.88
1000000	40	1.00
1000000	45	1.12
1000000	50	1.24
1500000	35	.72
1500000	40	.81
1500000	45	.91
1500000	50	1.01
2000000	35	.62
2000000	40	.70
2000000	45	.79
2000000	50	.87
2500000	35	.55
2500000	40	.63
2500000	45	.71
2500000	50	.78
3000000	35	.50
3000000	40	.57
3000000	45	.64
3000000	50	.71
3500000	35	.47
3500000	40	.53
3500000	45	.60
3500000	50	.66
4000000	35	.44
4000000	40	.50
4000000	45	.56
4000000	50	.62

DESIGN CONDITIONS: WIND= 5 MPH; TEMP.= 20 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.13
500000	40	1.29
500000	45	1.44
500000	50	1.60
1000000	35	.80
1000000	40	.91
1000000	45	1.02
1000000	50	1.13
1500000	35	.65
1500000	40	.74
1500000	45	.83
1500000	50	.92
2000000	35	.56
2000000	40	.64
2000000	45	.72
2000000	50	.80
2500000	35	.50
2500000	40	.57
2500000	45	.64
2500000	50	.71
3000000	35	.46
3000000	40	.52
3000000	45	.59
3000000	50	.65
3500000	35	.43
3500000	40	.48
3500000	45	.54
3500000	50	.60
4000000	35	.40
4000000	40	.45
4000000	45	.51
4000000	50	.56

DESIGN CONDITIONS: WIND= 5 MPH; TEMP.= 3Ø DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
5ØØØØØ	35	1.Ø2
5ØØØØØ	4Ø	1.15
5ØØØØØ	45	1.29
5ØØØØØ	5Ø	1.42
1ØØØØØØ	35	.72
1ØØØØØØ	4Ø	.81
1ØØØØØØ	45	.91
1ØØØØØØ	5Ø	1.ØØ
15ØØØØØ	35	.58
15ØØØØØ	4Ø	.66
15ØØØØØ	45	.74
15ØØØØØ	5Ø	.82
2ØØØØØØ	35	.51
2ØØØØØØ	4Ø	.57
2ØØØØØØ	45	.64
2ØØØØØØ	5Ø	.71
25ØØØØØ	35	.45
25ØØØØØ	4Ø	.51
25ØØØØØ	45	.57
25ØØØØØ	5Ø	.63
3ØØØØØØ	35	.41
3ØØØØØØ	4Ø	.47
3ØØØØØØ	45	.52
3ØØØØØØ	5Ø	.58
35ØØØØØ	35	.38
35ØØØØØ	4Ø	.43
35ØØØØØ	45	.48
35ØØØØØ	5Ø	.53
4ØØØØØØ	35	.36
4ØØØØØØ	4Ø	.4Ø
4ØØØØØØ	45	.45
4ØØØØØØ	5Ø	.5Ø

DESIGN CONDITIONS: WIND= 10 MPH; TEMP.=-10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.69
500000	40	1.91
500000	45	2.13
500000	50	2.34
1000000	35	1.19
1000000	40	1.35
1000000	45	1.50
1000000	50	1.65
1500000	35	.97
1500000	40	1.10
1500000	45	1.23
1500000	50	1.35
2000000	35	.84
2000000	40	.95
2000000	45	1.06
2000000	50	1.17
2500000	35	.75
2500000	40	.85
2500000	45	.95
2500000	50	1.04
3000000	35	.69
3000000	40	.78
3000000	45	.87
3000000	50	.95
3500000	35	.64
3500000	40	.72
3500000	45	.80
3500000	50	.88
4000000	35	.59
4000000	40	.67
4000000	45	.75
4000000	50	.82

DESIGN CONDITIONS: WIND= 10 MPH; TEMP.= 0 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.61
500000	40	1.82
500000	45	2.02
500000	50	2.23
1000000	35	1.14
1000000	40	1.28
1000000	45	1.43
1000000	50	1.57
1500000	35	.93
1500000	40	1.05
1500000	45	1.17
1500000	50	1.28
2000000	35	.80
2000000	40	.91
2000000	45	1.01
2000000	50	1.11
2500000	35	.72
2500000	40	.81
2500000	45	.90
2500000	50	.99
3000000	35	.66
3000000	40	.74
3000000	45	.82
3000000	50	.91
3500000	35	.61
3500000	40	.68
3500000	45	.76
3500000	50	.84
4000000	35	.57
4000000	40	.64
4000000	45	.71
4000000	50	.78

DESIGN CONDITIONS: WIND= 1Ø MPH; TEMP.= 1Ø DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
5ØØØØØ	35	1.53
5ØØØØØ	4Ø	1.72
5ØØØØØ	45	1.92
5ØØØØØ	5Ø	2.11
1ØØØØØØ	35	1.Ø8
1ØØØØØØ	4Ø	1.22
1ØØØØØØ	45	1.35
1ØØØØØØ	5Ø	1.49
15ØØØØØ	35	.88
15ØØØØØ	4Ø	.99
15ØØØØØ	45	1.1Ø
15ØØØØØ	5Ø	1.21
2ØØØØØØ	35	.76
2ØØØØØØ	4Ø	.86
2ØØØØØØ	45	.96
2ØØØØØØ	5Ø	1.Ø5
25ØØØØØ	35	.68
25ØØØØØ	4Ø	.77
25ØØØØØ	45	.85
25ØØØØØ	5Ø	.94
3ØØØØØØ	35	.62
3ØØØØØØ	4Ø	.7Ø
3ØØØØØØ	45	.78
3ØØØØØØ	5Ø	.86
35ØØØØØ	35	.58
35ØØØØØ	4Ø	.65
35ØØØØØ	45	.72
35ØØØØØ	5Ø	.79
4ØØØØØØ	35	.54
4ØØØØØØ	4Ø	.61
4ØØØØØØ	45	.67
4ØØØØØØ	5Ø	.74

DESIGN CONDITIONS: WIND= 10 MPH: TEMP.= 20 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.44
500000	40	1.62
500000	45	1.80
500000	50	1.98
1000000	35	1.02
1000000	40	1.15
1000000	45	1.27
1000000	50	1.40
1500000	35	.83
1500000	40	.94
1500000	45	1.04
1500000	50	1.14
2000000	35	.72
2000000	40	.81
2000000	45	.90
2000000	50	.99
2500000	35	.64
2500000	40	.72
2500000	45	.80
2500000	50	.88
3000000	35	.59
3000000	40	.66
3000000	45	.73
3000000	50	.80
3500000	35	.54
3500000	40	.61
3500000	45	.68
3500000	50	.74
4000000	35	.51
4000000	40	.57
4000000	45	.63
4000000	50	.70

DESIGN CONDITIONS: WIND= 10 MPH; TEMP.= 30 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.35
500000	40	1.52
500000	45	1.68
500000	50	1.84
1000000	35	.95
1000000	40	1.07
1000000	45	1.19
1000000	50	1.30
1500000	35	.78
1500000	40	.87
1500000	45	.97
1500000	50	1.06
2000000	35	.67
2000000	40	.76
2000000	45	.84
2000000	50	.92
2500000	35	.60
2500000	40	.68
2500000	45	.75
2500000	50	.82
3000000	35	.55
3000000	40	.62
3000000	45	.68
3000000	50	.75
3500000	35	.51
3500000	40	.57
3500000	45	.63
3500000	50	.69
4000000	35	.47
4000000	40	.53
4000000	45	.59
4000000	50	.65

DESIGN CONDITIONS: WIND= 15 MPH; TEMP.= -10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	2.05
500000	40	2.30
500000	45	2.54
500000	50	2.79
1000000	35	1.45
1000000	40	1.62
1000000	45	1.80
1000000	50	1.97
1500000	35	1.18
1500000	40	1.32
1500000	45	1.47
1500000	50	1.61
2000000	35	1.02
2000000	40	1.15
2000000	45	1.27
2000000	50	1.39
2500000	35	.91
2500000	40	1.02
2500000	45	1.13
2500000	50	1.24
3000000	35	.83
3000000	40	.93
3000000	45	1.03
3000000	50	1.13
3500000	35	.77
3500000	40	.86
3500000	45	.96
3500000	50	1.05
4000000	35	.72
4000000	40	.81
4000000	45	.90
4000000	50	.98

DESIGN CONDITIONS: WIND= 15 MPH; TEMP.= 0 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.98
500000	40	2.22
500000	45	2.46
500000	50	2.69
1000000	35	1.40
1000000	40	1.57
1000000	45	1.74
1000000	50	1.90
1500000	35	1.14
1500000	40	1.28
1500000	45	1.42
1500000	50	1.55
2000000	35	.99
2000000	40	1.11
2000000	45	1.23
2000000	50	1.34
2500000	35	.88
2500000	40	.99
2500000	45	1.10
2500000	50	1.20
3000000	35	.81
3000000	40	.90
3000000	45	1.00
3000000	50	1.10
3500000	35	.75
3500000	40	.84
3500000	45	.93
3500000	50	1.01
4000000	35	.70
4000000	40	.78
4000000	45	.87
4000000	50	.95

DESIGN CONDITIONS: WIND= 15 MPH; TEMP.= 10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.92
500000	40	2.14
500000	45	2.37
500000	50	2.59
1000000	35	1.35
1000000	40	1.51
1000000	45	1.67
1000000	50	1.83
1500000	35	1.10
1500000	40	1.24
1500000	45	1.37
1500000	50	1.49
2000000	35	.96
2000000	40	1.07
2000000	45	1.18
2000000	50	1.29
2500000	35	.85
2500000	40	.96
2500000	45	1.06
2500000	50	1.16
3000000	35	.78
3000000	40	.87
3000000	45	.96
3000000	50	1.05
3500000	35	.72
3500000	40	.81
3500000	45	.89
3500000	50	.98
4000000	35	.67
4000000	40	.75
4000000	45	.83
4000000	50	.91

DESIGN CONDITIONS: WIND= 15 MPH; TEMP.= 20 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.85
500000	40	2.06
500000	45	2.28
500000	50	2.49
1000000	35	1.31
1000000	40	1.46
1000000	45	1.61
1000000	50	1.76
1500000	35	1.06
1500000	40	1.19
1500000	45	1.31
1500000	50	1.43
2000000	35	.92
2000000	40	1.03
2000000	45	1.14
2000000	50	1.24
2500000	35	.82
2500000	40	.92
2500000	45	1.02
2500000	50	1.11
3000000	35	.75
3000000	40	.84
3000000	45	.93
3000000	50	1.01
3500000	35	.70
3500000	40	.78
3500000	45	.86
3500000	50	.94
4000000	35	.65
4000000	40	.73
4000000	45	.80
4000000	50	.88

DESIGN CONDITIONS: WIND= 15 MPH; TEMP.= 30 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	1.78
500000	40	1.98
500000	45	2.18
500000	50	2.38
1000000	35	1.26
1000000	40	1.40
1000000	45	1.54
1000000	50	1.68
1500000	35	1.02
1500000	40	1.14
1500000	45	1.26
1500000	50	1.37
2000000	35	.89
2000000	40	.99
2000000	45	1.09
2000000	50	1.19
2500000	35	.79
2500000	40	.88
2500000	45	.97
2500000	50	1.06
3000000	35	.72
3000000	40	.81
3000000	45	.89
3000000	50	.97
3500000	35	.67
3500000	40	.75
3500000	45	.82
3500000	50	.90
4000000	35	.63
4000000	40	.70
4000000	45	.77
4000000	50	.84

DESIGN CONDITIONS: WIND= 20 MPH; TEMP.=-10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	2.46
500000	40	2.75
500000	45	3.03
500000	50	3.31
1000000	35	1.74
1000000	40	1.94
1000000	45	2.14
1000000	50	2.34
1500000	35	1.42
1500000	40	1.58
1500000	45	1.75
1500000	50	1.91
2000000	35	1.23
2000000	40	1.37
2000000	45	1.51
2000000	50	1.65
2500000	35	1.10
2500000	40	1.23
2500000	45	1.35
2500000	50	1.48
3000000	35	1.00
3000000	40	1.12
3000000	45	1.23
3000000	50	1.35
3500000	35	.93
3500000	40	1.04
3500000	45	1.14
3500000	50	1.25
4000000	35	.87
4000000	40	.97
4000000	45	1.07
4000000	50	1.17

DESIGN CONDITIONS: WIND= 20 MPH; TEMP.= 0 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	2.41
500000	40	2.69
500000	45	2.96
500000	50	3.23
1000000	35	1.70
1000000	40	1.90
1000000	45	2.09
1000000	50	2.28
1500000	35	1.39
1500000	40	1.55
1500000	45	1.71
1500000	50	1.86
2000000	35	1.20
2000000	40	1.34
2000000	45	1.48
2000000	50	1.61
2500000	35	1.07
2500000	40	1.20
2500000	45	1.32
2500000	50	1.44
3000000	35	.98
3000000	40	1.09
3000000	45	1.21
3000000	50	1.31
3500000	35	.91
3500000	40	1.01
3500000	45	1.12
3500000	50	1.22
4000000	35	.85
4000000	40	.95
4000000	45	1.04
4000000	50	1.14

DESIGN CONDITIONS: WIND= 20 MPH; TEMP.= 10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	2.35
500000	40	2.62
500000	45	2.89
500000	50	3.15
1000000	35	1.66
1000000	40	1.85
1000000	45	2.04
1000000	50	2.22
1500000	35	1.36
1500000	40	1.51
1500000	45	1.66
1500000	50	1.81
2000000	35	1.17
2000000	40	1.31
2000000	45	1.44
2000000	50	1.57
2500000	35	1.05
2500000	40	1.17
2500000	45	1.29
2500000	50	1.40
3000000	35	.96
3000000	40	1.07
3000000	45	1.18
3000000	50	1.28
3500000	35	.89
3500000	40	.99
3500000	45	1.09
3500000	50	1.19
4000000	35	.83
4000000	40	.92
4000000	45	1.02
4000000	50	1.11

DESIGN CONDITIONS: WIND= 20 MPH; TEMP.= 20 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	2.30
500000	40	2.56
500000	45	2.81
500000	50	3.06
1000000	35	1.62
1000000	40	1.81
1000000	45	1.99
1000000	50	2.16
1500000	35	1.32
1500000	40	1.47
1500000	45	1.62
1500000	50	1.76
2000000	35	1.15
2000000	40	1.28
2000000	45	1.40
2000000	50	1.53
2500000	35	1.02
2500000	40	1.14
2500000	45	1.25
2500000	50	1.37
3000000	35	.94
3000000	40	1.04
3000000	45	1.14
3000000	50	1.25
3500000	35	.87
3500000	40	.96
3500000	45	1.06
3500000	50	1.15
4000000	35	.81
4000000	40	.90
4000000	45	.99
4000000	50	1.08

DESIGN CONDITIONS: WIND= 20 MPH; TEMP.= 30 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	2.24
500000	40	2.49
500000	45	2.73
500000	50	2.97
1000000	35	1.58
1000000	40	1.76
1000000	45	1.93
1000000	50	2.10
1500000	35	1.29
1500000	40	1.44
1500000	45	1.58
1500000	50	1.71
2000000	35	1.12
2000000	40	1.24
2000000	45	1.36
2000000	50	1.48
2500000	35	1.00
2500000	40	1.11
2500000	45	1.22
2500000	50	1.33
3000000	35	.91
3000000	40	1.01
3000000	45	1.11
3000000	50	1.21
3500000	35	.84
3500000	40	.94
3500000	45	1.03
3500000	50	1.12
4000000	35	.79
4000000	40	.88
4000000	45	.96
4000000	50	1.05

DESIGN CONDITIONS: WIND= 25 MPH; TEMP.=-10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	2.91
500000	40	3.24
500000	45	3.56
500000	50	3.88
1000000	35	2.05
1000000	40	2.29
1000000	45	2.52
1000000	50	2.74
1500000	35	1.68
1500000	40	1.87
1500000	45	2.05
1500000	50	2.24
2000000	35	1.45
2000000	40	1.62
2000000	45	1.78
2000000	50	1.94
2500000	35	1.30
2500000	40	1.44
2500000	45	1.59
2500000	50	1.73
3000000	35	1.18
3000000	40	1.32
3000000	45	1.45
3000000	50	1.58
3500000	35	1.10
3500000	40	1.22
3500000	45	1.34
3500000	50	1.46
4000000	35	1.02
4000000	40	1.14
4000000	45	1.26
4000000	50	1.37

DESIGN CONDITIONS: WIND= 25 MPH; TEMP.= 0 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	2.86
500000	40	3.18
500000	45	3.50
500000	50	3.81
1000000	35	2.02
1000000	40	2.25
1000000	45	2.47
1000000	50	2.69
1500000	35	1.65
1500000	40	1.84
1500000	45	2.02
1500000	50	2.20
2000000	35	1.43
2000000	40	1.59
2000000	45	1.75
2000000	50	1.90
2500000	35	1.28
2500000	40	1.42
2500000	45	1.56
2500000	50	1.70
3000000	35	1.17
3000000	40	1.30
3000000	45	1.43
3000000	50	1.55
3500000	35	1.08
3500000	40	1.20
3500000	45	1.32
3500000	50	1.44
4000000	35	1.01
4000000	40	1.12
4000000	45	1.23
4000000	50	1.34

DESIGN CONDITIONS: WIND= 25 MPH; TEMP.= 10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	2.82
500000	40	3.13
500000	45	3.44
500000	50	3.74
1000000	35	1.99
1000000	40	2.21
1000000	45	2.43
1000000	50	2.64
1500000	35	1.62
1500000	40	1.81
1500000	45	1.98
1500000	50	2.16
2000000	35	1.41
2000000	40	1.56
2000000	45	1.72
2000000	50	1.87
2500000	35	1.26
2500000	40	1.40
2500000	45	1.54
2500000	50	1.67
3000000	35	1.15
3000000	40	1.28
3000000	45	1.40
3000000	50	1.52
3500000	35	1.06
3500000	40	1.18
3500000	45	1.30
3500000	50	1.41
4000000	35	.99
4000000	40	1.10
4000000	45	1.21
4000000	50	1.32

DESIGN CONDITIONS: WIND= 25 MPH; TEMP.= 20 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	2.77
500000	40	3.08
500000	45	3.38
500000	50	3.67
1000000	35	1.96
1000000	40	2.17
1000000	45	2.39
1000000	50	2.59
1500000	35	1.60
1500000	40	1.77
1500000	45	1.95
1500000	50	2.12
2000000	35	1.38
2000000	40	1.54
2000000	45	1.69
2000000	50	1.83
2500000	35	1.24
2500000	40	1.37
2500000	45	1.51
2500000	50	1.64
3000000	35	1.13
3000000	40	1.25
3000000	45	1.38
3000000	50	1.49
3500000	35	1.04
3500000	40	1.16
3500000	45	1.27
3500000	50	1.38
4000000	35	.98
4000000	40	1.08
4000000	45	1.19
4000000	50	1.29

DESIGN CONDITIONS: WIND= 25 MPH; TEMP.= 30 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	2.72
500000	40	3.02
500000	45	3.31
500000	50	3.60
1000000	35	1.92
1000000	40	2.14
1000000	45	2.34
1000000	50	2.54
1500000	35	1.57
1500000	40	1.74
1500000	45	1.91
1500000	50	2.07
2000000	35	1.36
2000000	40	1.51
2000000	45	1.65
2000000	50	1.80
2500000	35	1.22
2500000	40	1.35
2500000	45	1.48
2500000	50	1.61
3000000	35	1.11
3000000	40	1.23
3000000	45	1.35
3000000	50	1.47
3500000	35	1.03
3500000	40	1.14
3500000	45	1.25
3500000	50	1.36
4000000	35	.96
4000000	40	1.07
4000000	45	1.17
4000000	50	1.27

DESIGN CONDITIONS: WIND= 30 MPH; TEMP.=-10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	3.37
500000	40	3.75
500000	45	4.12
500000	50	4.48
1000000	35	2.38
1000000	40	2.65
1000000	45	2.91
1000000	50	3.16
1500000	35	1.95
1500000	40	2.16
1500000	45	2.38
1500000	50	2.58
2000000	35	1.68
2000000	40	1.87
2000000	45	2.06
2000000	50	2.24
2500000	35	1.51
2500000	40	1.67
2500000	45	1.84
2500000	50	2.00
3000000	35	1.37
3000000	40	1.53
3000000	45	1.68
3000000	50	1.82
3500000	35	1.27
3500000	40	1.41
3500000	45	1.55
3500000	50	1.69
4000000	35	1.19
4000000	40	1.32
4000000	45	1.45
4000000	50	1.58

DESIGN CONDITIONS: WIND= 30 MPH; TEMP.= 0 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	3.34
500000	40	3.70
500000	45	4.07
500000	50	4.42
1000000	35	2.36
1000000	40	2.62
1000000	45	2.87
1000000	50	3.12
1500000	35	1.92
1500000	40	2.14
1500000	45	2.34
1500000	50	2.55
2000000	35	1.67
2000000	40	1.85
2000000	45	2.03
2000000	50	2.21
2500000	35	1.49
2500000	40	1.65
2500000	45	1.82
2500000	50	1.97
3000000	35	1.36
3000000	40	1.51
3000000	45	1.66
3000000	50	1.80
3500000	35	1.26
3500000	40	1.40
3500000	45	1.53
3500000	50	1.67
4000000	35	1.18
4000000	40	1.31
4000000	45	1.43
4000000	50	1.56

DESIGN CONDITIONS: WIND= 30 MPH; TEMP.= 10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	3.30
500000	40	3.66
500000	45	4.01
500000	50	4.36
1000000	35	2.33
1000000	40	2.59
1000000	45	2.84
1000000	50	3.08
1500000	35	1.90
1500000	40	2.11
1500000	45	2.31
1500000	50	2.51
2000000	35	1.65
2000000	40	1.83
2000000	45	2.00
2000000	50	2.18
2500000	35	1.47
2500000	40	1.63
2500000	45	1.79
2500000	50	1.95
3000000	35	1.34
3000000	40	1.49
3000000	45	1.64
3000000	50	1.78
3500000	35	1.24
3500000	40	1.38
3500000	45	1.51
3500000	50	1.64
4000000	35	1.16
4000000	40	1.29
4000000	45	1.42
4000000	50	1.54

DESIGN CONDITIONS: WIND= 30 MPH; TEMP.= 20 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	3.26
500000	40	3.61
500000	45	3.96
500000	50	4.30
1000000	35	2.30
1000000	40	2.55
1000000	45	2.80
1000000	50	3.04
1500000	35	1.88
1500000	40	2.08
1500000	45	2.28
1500000	50	2.48
2000000	35	1.63
2000000	40	1.80
2000000	45	1.98
2000000	50	2.15
2500000	35	1.45
2500000	40	1.61
2500000	45	1.77
2500000	50	1.92
3000000	35	1.33
3000000	40	1.47
3000000	45	1.61
3000000	50	1.75
3500000	35	1.23
3500000	40	1.36
3500000	45	1.49
3500000	50	1.62
4000000	35	1.15
4000000	40	1.27
4000000	45	1.40
4000000	50	1.52

DESIGN CONDITIONS: WIND= 30 MPH; TEMP.= 30 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	3.22
500000	40	3.57
500000	45	3.90
500000	50	4.24
1000000	35	2.27
1000000	40	2.52
1000000	45	2.76
1000000	50	2.99
1500000	35	1.86
1500000	40	2.06
1500000	45	2.25
1500000	50	2.44
2000000	35	1.61
2000000	40	1.78
2000000	45	1.95
2000000	50	2.12
2500000	35	1.44
2500000	40	1.59
2500000	45	1.74
2500000	50	1.89
3000000	35	1.31
3000000	40	1.45
3000000	45	1.59
3000000	50	1.73
3500000	35	1.21
3500000	40	1.34
3500000	45	1.47
3500000	50	1.60
4000000	35	1.13
4000000	40	1.26
4000000	45	1.38
4000000	50	1.49

DESIGN CONDITIONS: WIND= 35 MPH; TEMP.=-10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	3.85
500000	40	4.28
500000	45	4.69
500000	50	5.10
1000000	35	2.72
1000000	40	3.02
1000000	45	3.32
1000000	50	3.60
1500000	35	2.22
1500000	40	2.47
1500000	45	2.71
1500000	50	2.94
2000000	35	1.92
2000000	40	2.14
2000000	45	2.34
2000000	50	2.55
2500000	35	1.72
2500000	40	1.91
2500000	45	2.10
2500000	50	2.28
3000000	35	1.57
3000000	40	1.74
3000000	45	1.91
3000000	50	2.08
3500000	35	1.45
3500000	40	1.61
3500000	45	1.77
3500000	50	1.92
4000000	35	1.36
4000000	40	1.51
4000000	45	1.66
4000000	50	1.80

DESIGN CONDITIONS: WIND= 35 MPH; TEMP.= 0 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	3.82
500000	40	4.24
500000	45	4.65
500000	50	5.04
1000000	35	2.70
1000000	40	3.00
1000000	45	3.28
1000000	50	3.56
1500000	35	2.20
1500000	40	2.44
1500000	45	2.68
1500000	50	2.91
2000000	35	1.91
2000000	40	2.12
2000000	45	2.32
2000000	50	2.52
2500000	35	1.71
2500000	40	1.89
2500000	45	2.07
2500000	50	2.25
3000000	35	1.56
3000000	40	1.73
3000000	45	1.89
3000000	50	2.06
3500000	35	1.44
3500000	40	1.60
3500000	45	1.75
3500000	50	1.90
4000000	35	1.35
4000000	40	1.50
4000000	45	1.64
4000000	50	1.78

DESIGN CONDITIONS: WIND= 35 MPH; TEMP.= 10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	3.79
500000	40	4.20
500000	45	4.60
500000	50	4.99
1000000	35	2.68
1000000	40	2.97
1000000	45	3.25
1000000	50	3.53
1500000	35	2.18
1500000	40	2.42
1500000	45	2.65
1500000	50	2.88
2000000	35	1.89
2000000	40	2.10
2000000	45	2.30
2000000	50	2.49
2500000	35	1.69
2500000	40	1.87
2500000	45	2.05
2500000	50	2.23
3000000	35	1.54
3000000	40	1.71
3000000	45	1.87
3000000	50	2.03
3500000	35	1.43
3500000	40	1.58
3500000	45	1.74
3500000	50	1.88
4000000	35	1.34
4000000	40	1.48
4000000	45	1.62
4000000	50	1.76

DESIGN CONDITIONS: WIND= 35 MPH; TEMP.= 20 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	3.75
500000	40	4.16
500000	45	4.55
500000	50	4.94
1000000	35	2.65
1000000	40	2.94
1000000	45	3.22
1000000	50	3.49
1500000	35	2.16
1500000	40	2.40
1500000	45	2.63
1500000	50	2.85
2000000	35	1.87
2000000	40	2.08
2000000	45	2.27
2000000	50	2.47
2500000	35	1.67
2500000	40	1.86
2500000	45	2.03
2500000	50	2.21
3000000	35	1.53
3000000	40	1.69
3000000	45	1.86
3000000	50	2.01
3500000	35	1.41
3500000	40	1.57
3500000	45	1.72
3500000	50	1.86
4000000	35	1.32
4000000	40	1.47
4000000	45	1.61
4000000	50	1.74

DESIGN CONDITIONS: WIND= 35 MPH; TEMP.= 30 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	3.72
500000	40	4.12
500000	45	4.51
500000	50	4.88
1000000	35	2.63
1000000	40	2.91
1000000	45	3.18
1000000	50	3.45
1500000	35	2.14
1500000	40	2.37
1500000	45	2.60
1500000	50	2.82
2000000	35	1.86
2000000	40	2.06
2000000	45	2.25
2000000	50	2.44
2500000	35	1.66
2500000	40	1.84
2500000	45	2.01
2500000	50	2.18
3000000	35	1.51
3000000	40	1.68
3000000	45	1.84
3000000	50	1.99
3500000	35	1.40
3500000	40	1.55
3500000	45	1.70
3500000	50	1.84
4000000	35	1.31
4000000	40	1.45
4000000	45	1.59
4000000	50	1.72

DESIGN CONDITIONS: WIND= 40 MPH; TEMP.=-10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	4.34
500000	40	4.81
500000	45	5.28
500000	50	5.73
1000000	35	3.07
1000000	40	3.40
1000000	45	3.73
1000000	50	4.05
1500000	35	2.50
1500000	40	2.78
1500000	45	3.04
1500000	50	3.30
2000000	35	2.17
2000000	40	2.40
2000000	45	2.64
2000000	50	2.86
2500000	35	1.94
2500000	40	2.15
2500000	45	2.36
2500000	50	2.56
3000000	35	1.77
3000000	40	1.96
3000000	45	2.15
3000000	50	2.33
3500000	35	1.64
3500000	40	1.82
3500000	45	1.99
3500000	50	2.16
4000000	35	1.53
4000000	40	1.70
4000000	45	1.86
4000000	50	2.02

DESIGN CONDITIONS: WIND= 40 MPH; TEMP.= 0 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	4.31
500000	40	4.78
500000	45	5.24
500000	50	5.68
1000000	35	3.05
1000000	40	3.38
1000000	45	3.70
1000000	50	4.02
1500000	35	2.49
1500000	40	2.76
1500000	45	3.02
1500000	50	3.28
2000000	35	2.15
2000000	40	2.39
2000000	45	2.62
2000000	50	2.84
2500000	35	1.93
2500000	40	2.13
2500000	45	2.34
2500000	50	2.54
3000000	35	1.76
3000000	40	1.95
3000000	45	2.13
3000000	50	2.32
3500000	35	1.63
3500000	40	1.80
3500000	45	1.98
3500000	50	2.14
4000000	35	1.52
4000000	40	1.69
4000000	45	1.85
4000000	50	2.01

DESIGN CONDITIONS: WIND= 40 MPH; TEMP.= 10 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	4.28
500000	40	4.74
500000	45	5.19
500000	50	5.63
1000000	35	3.03
1000000	40	3.35
1000000	45	3.67
1000000	50	3.98
1500000	35	2.47
1500000	40	2.74
1500000	45	3.00
1500000	50	3.25
2000000	35	2.14
2000000	40	2.37
2000000	45	2.59
2000000	50	2.81
2500000	35	1.91
2500000	40	2.12
2500000	45	2.32
2500000	50	2.52
3000000	35	1.74
3000000	40	1.93
3000000	45	2.12
3000000	50	2.30
3500000	35	1.62
3500000	40	1.79
3500000	45	1.96
3500000	50	2.13
4000000	35	1.51
4000000	40	1.67
4000000	45	1.83
4000000	50	1.99

DESIGN CONDITIONS: WIND= 40 MPH; TEMP.= 20 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	4.25
500000	40	4.71
500000	45	5.15
500000	50	5.59
1000000	35	3.00
1000000	40	3.33
1000000	45	3.64
1000000	50	3.95
1500000	35	2.45
1500000	40	2.72
1500000	45	2.97
1500000	50	3.22
2000000	35	2.12
2000000	40	2.35
2000000	45	2.57
2000000	50	2.79
2500000	35	1.90
2500000	40	2.10
2500000	45	2.30
2500000	50	2.50
3000000	35	1.73
3000000	40	1.92
3000000	45	2.10
3000000	50	2.28
3500000	35	1.60
3500000	40	1.78
3500000	45	1.94
3500000	50	2.11
4000000	35	1.50
4000000	40	1.66
4000000	45	1.82
4000000	50	1.97

DESIGN CONDITIONS: WIND= 40 MPH; TEMP.= 30 DEGREES F.

VOLUME, CUBIC FEET	HEIGHT, FEET	AIR CHANGES/HOUR
500000	35	4.22
500000	40	4.67
500000	45	5.11
500000	50	5.54
1000000	35	2.98
1000000	40	3.30
1000000	45	3.61
1000000	50	3.92
1500000	35	2.43
1500000	40	2.70
1500000	45	2.95
1500000	50	3.20
2000000	35	2.11
2000000	40	2.33
2000000	45	2.55
2000000	50	2.77
2500000	35	1.88
2500000	40	2.09
2500000	45	2.28
2500000	50	2.47
3000000	35	1.72
3000000	40	1.90
3000000	45	2.08
3000000	50	2.26
3500000	35	1.59
3500000	40	1.76
3500000	45	1.93
3500000	50	2.09
4000000	35	1.49
4000000	40	1.65
4000000	45	1.80
4000000	50	1.96

Appendix B

NCEL COLD JET DESTRATIFIER DESIGN TABLE

HANGAR VOLUME: .5 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
70	4	2000	1570	15.28
80	4	2000	2050	13.38
90	4	2000	2590	11.90
100	4	2000	3200	10.71
110	4	2000	3870	9.74

HANGAR VOLUME: .75 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
80	6	2000	2050	13.38
90	6	2000	2590	11.90
100	6	2000	3200	10.71
110	6	2000	3870	9.74
120	6	2000	4610	8.92
130	6	2000	5410	8.23
140	6	2000	6280	7.64

HANGAR VOLUME: 1 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
90	6	3000	1730	17.83
100	6	3000	2130	16.07
110	6	3000	2580	14.60
120	6	3000	3070	13.39
130	6	3000	3610	12.34
140	6	3000	4180	11.47
150	6	3000	4800	10.71
160	6	3000	5460	10.04

HANGAR VOLUME: 1.25 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
110	6	3500	2210	17.04
120	6	3500	2630	15.62
130	6	3500	3090	14.41
140	6	3500	3580	13.39
150	6	3500	4120	12.48
160	6	3500	4680	11.71
170	6	3500	5290	11.01
180	6	3500	5930	10.40

HANGAR VOLUME: 1.5 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
120	6	4500	2050	20.06
130	6	4500	2400	18.54
140	6	4500	2790	17.20
150	6	4500	3200	16.06
160	6	4500	3640	15.06
170	6	4500	4110	14.17
180	6	4500	4610	13.38
190	6	4500	5140	12.67
200	6	4500	5690	12.04

HANGAR VOLUME: 1.25 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
110	6	3500	2210	17.04
120	6	3500	2630	15.62
130	6	3500	3090	14.41
140	6	3500	3580	13.39
150	6	3500	4120	12.48
160	6	3500	4680	11.71
170	6	3500	5290	11.01
180	6	3500	5930	10.40

HANGAR VOLUME: 1.5 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
120	6	4500	2050	20.06
130	6	4500	2400	18.54
140	6	4500	2790	17.20
150	6	4500	3200	16.06
160	6	4500	3640	15.06
170	6	4500	4110	14.17
180	6	4500	4610	13.38
190	6	4500	5140	12.67
200	6	4500	5690	12.04

HANGAR VOLUME: 1.75 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
130	6	5000	2160	20.60
140	6	5000	2510	19.11
150	6	5000	2880	17.84
160	6	5000	3280	16.72
170	6	5000	3700	15.74
180	6	5000	4150	14.86
190	6	5000	4620	14.09
200	6	5000	5120	13.38
210	6	5000	5650	12.74
220	6	5000	6200	12.16

HANGAR VOLUME: 2 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
140	6	5500	2280	21.03
150	6	5500	2620	19.62
160	6	5500	2980	18.40
170	6	5500	3360	17.33
180	6	5500	3770	16.36
190	6	5500	4200	15.50
200	6	5500	4660	14.71
210	6	5500	5130	14.02
220	6	5500	5640	13.37
230	6	5500	6160	12.80

HANGAR VOLUME: 2.25 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
140	6	6500	1930	24.85
150	6	6500	2210	23.22
160	6	6500	2520	21.75
170	6	6500	2840	20.49
180	6	6500	3190	19.33
190	6	6500	3560	18.30
200	6	6500	3940	17.39
210	6	6500	4340	16.57
220	6	6500	4770	15.81
230	6	6500	5210	15.13
240	6	6500	5680	14.49
250	6	6500	6160	13.91

HANGAR VOLUME: 2.5 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
150	6	7000	2060	24.96
160	6	7000	2340	23.42
170	6	7000	2640	22.05
180	6	7000	2960	20.82
190	6	7000	3300	19.72
200	6	7000	3660	18.73
210	6	7000	4030	17.85
220	6	7000	4430	17.02
230	6	7000	4840	16.29
240	6	7000	5270	15.61
250	6	7000	5720	14.98
260	6	7000	6190	14.40

HANGAR VOLUME: 2.75 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
160	6	8000	2050	26.75
170	6	8000	2310	25.20
180	6	8000	2590	23.80
190	6	8000	2890	22.53
200	6	8000	3200	21.41
210	6	8000	3530	20.39
220	6	8000	3870	19.47
230	6	8000	4230	18.62
240	6	8000	4610	17.84
250	6	8000	5000	17.13
260	6	8000	5410	16.47
270	6	8000	5840	15.85
280	6	8000	6280	15.28

HANGAR VOLUME: 3 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
170	6	9000	2050	28.37
180	6	9000	2300	26.79
190	6	9000	2570	25.34
200	6	9000	2840	24.11
210	6	9000	3140	22.93
220	6	9000	3440	21.90
230	6	9000	3760	20.95
240	6	9000	4100	20.06
250	6	9000	4450	19.26
260	6	9000	4810	18.52
270	6	9000	5190	17.83
280	6	9000	5580	17.20
290	6	9000	5980	16.61

HANGAR VOLUME: 3.25 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
170	6	9000	2050	28.37
180	6	9000	2300	26.79
190	6	9000	2570	25.34
200	6	9000	2840	24.11
210	6	9000	3140	22.93
220	6	9000	3440	21.90
230	6	9000	3760	20.95
240	6	9000	4100	20.06
250	6	9000	4450	19.26
260	6	9000	4810	18.52
270	6	9000	5190	17.83
280	6	9000	5580	17.20
290	6	9000	5980	16.61
300	6	9000	6410	16.05

HANGAR VOLUME: 3.5 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
180	6	10000	2070	29.76
190	6	10000	2310	28.18
200	6	10000	2560	26.76
210	6	10000	2820	25.50
220	6	10000	3100	24.32
230	6	10000	3390	23.26
240	6	10000	3690	22.29
250	6	10000	4000	21.41
260	6	10000	4330	20.58
270	6	10000	4670	19.82
280	6	10000	5020	19.11
290	6	10000	5390	18.45
300	6	10000	5760	17.84
310	6	10000	6160	17.25

HANGAR VOLUME: 3.75 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
190	6	10500	2200	29.58
200	6	10500	2440	28.09
210	6	10500	2690	26.75
220	6	10500	2950	25.55
230	6	10500	3220	24.45
240	6	10500	3510	23.42
250	6	10500	3810	22.48
260	6	10500	4120	21.62
270	6	10500	4450	20.80
280	6	10500	4780	20.07
290	6	10500	5130	19.37
300	6	10500	5490	18.73
310	6	10500	5860	18.13
320	6	10500	6250	17.55

HANGAR VOLUME: 4 MILLION CUBIC FEET

DESTRATIFIER PARAMETERS

HANGAR WIDTH FEET	NUMBER UNITS	FLOW CFM	DISCHARGE VELOCITY FPM	NOZZLE DIAMETER INCHES
190	6	11000	2100	30.99
200	6	11000	2330	29.42
210	6	11000	2560	28.07
220	6	11000	2820	26.75
230	6	11000	3080	25.59
240	6	11000	3350	24.54
250	6	11000	3640	23.54
260	6	11000	3930	22.66
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